POWER

Vol. 44 NEW YORK, SEPTEMBER 19, 1916 No. 12

CANDID CHATS

Success

If you are ever a success it will be because you take hold of yourself and develop yourself and direct yourself in ways useful to the world about you.

The world pays nobody except for work it wants done. If you do better work than your neighbor you can command better pay. The world is not going to give you better pay unless you work for it, but if you never do good work 'til the world pays you high wages you will never get high pay.

You have to do such work that the world will wake up and take notice that you are the one who does that good work. You have to do this whether you receive pay for it or not. In other words, when you are beginning in any line of work you have to do good work for small pay or for nothing as a matter of advertisement of your work. Then when those around you begin to demand your work, you can fix your pay according to right principles. But if you are full of spite and contempt and ill feeling for those around you who have not paid you high prices for your work, you will cut off the connection between yourself and the good pay you want. All kinds of envy and criticism and fault-finding and ugly feeling demagnetize you so that you cannot attract the things you desire.

Good will to all—to the unjust as well as to the just is the magnetic power that enables you to attract the things you desire. If you fill yourself with good will, it is as if you turned the current onto the wires that do the work. If you fill yourself with envy and resentment and criticism and faultfinding of the world and the people around you, it is as if you turned off the good-will power which does the work you want done. Go into work and stick to it. Do good work and do it in good will to the world. As the demand for your services becomes greater, so will your value increase.

Then there is another side of life that must be systematized. No matter how small your income is, it must be managed so that there will be a little laid by for the day of opportunity. Allot your money for things that are necessary to buy, taking pains that your savings account is not left to the last, but is reckoned in with the very important things like paying your tailor and shoemaker, so that your savings account grows every day and every month if only by a twenty-five cent piece.

If you cast about you, you will see that even men in very poor circumstances indulge in a great many luxuries. Wastefulness is the greatest luxury in which they indulge. The poor coal passer will be more wasteful than the manager would be if he were in the same place, just because he is more careless and less determined to manage things at his command.

Work with a good will and be determined to advance, and your success is sure to come.

Contributed by A. A. NORRMAN, *Pottsville, Penn.*

Extension to West Penn Co.'s Connellsville Power Plant

By Warren O. Rogers

SYNOPSIS—The addition to the Connellsville, Penn., plant consists of three water-tube boilers, of 13,710 sq.ft. of surface each, and each equipped with a 14-retort underfeed stoker. The turbine that these boilers serve is of 18,000 kw. capacity. Between each boiler and its stack is an 8,240-sq.ft. economizer. Fuel is handled by a traveling crane with grab-bucket from railway cars direct to the stoker hoppers, or to a traveling crusher, when run-of-mine coal is received. The steam pressure carried is 200 lb. The turbines will run with nearly a constant load, the older portion of the plant carrying the fluctuations.

The constantly growing demand for electrical energy has necessitated the extension of the Connellsville, Penn., power plant of the West Penn Traction Co. The new boilers are in an addition to the old building. The new 18,000-kw. turbine is in the original turbine room. Although the new boilers and the turbine are to operate as one unit, the main steam line from them will be connected to that of the older boiler plant so that steam can be supplied from the boilers in either plant to the main steam header.

The new extension contains three boilers, Fig. 1, each having 13,710 ft. of heating surface. These boilers are built for a working pressure of 250 lb. and are constructed in accordance with the A. S. M. E. Boiler Code. They have no unusual features other than that they are much larger than most boilers of this type. They are set considerably higher than usual so as to provide proper furnace conditions and combustion volume, and the baffling is of the so-called four-pass type.

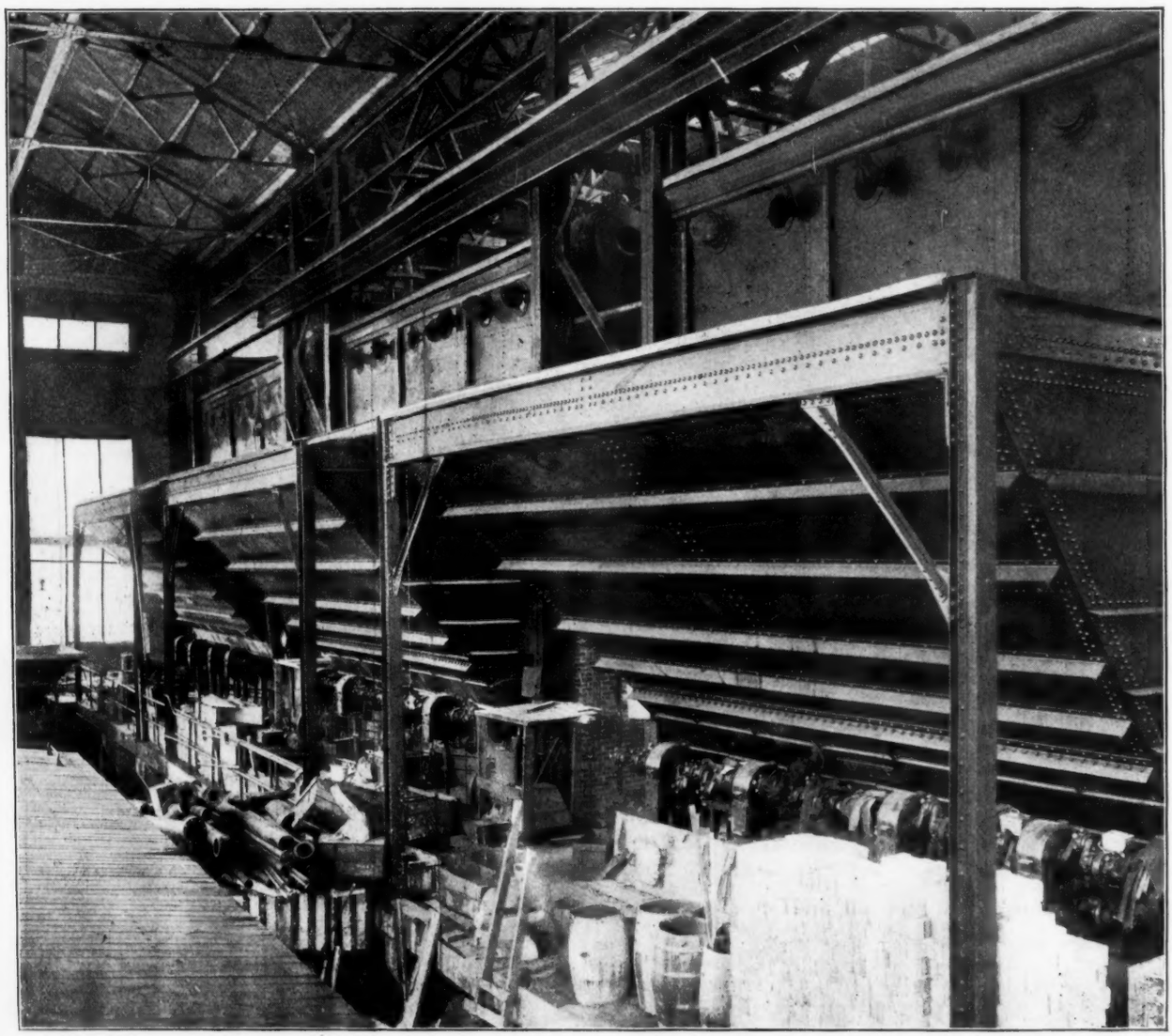


FIG. 1. THREE WATER-TUBE BOILERS, EACH CONTAINING 13,710 SQ.FT. OF HEATING SURFACE

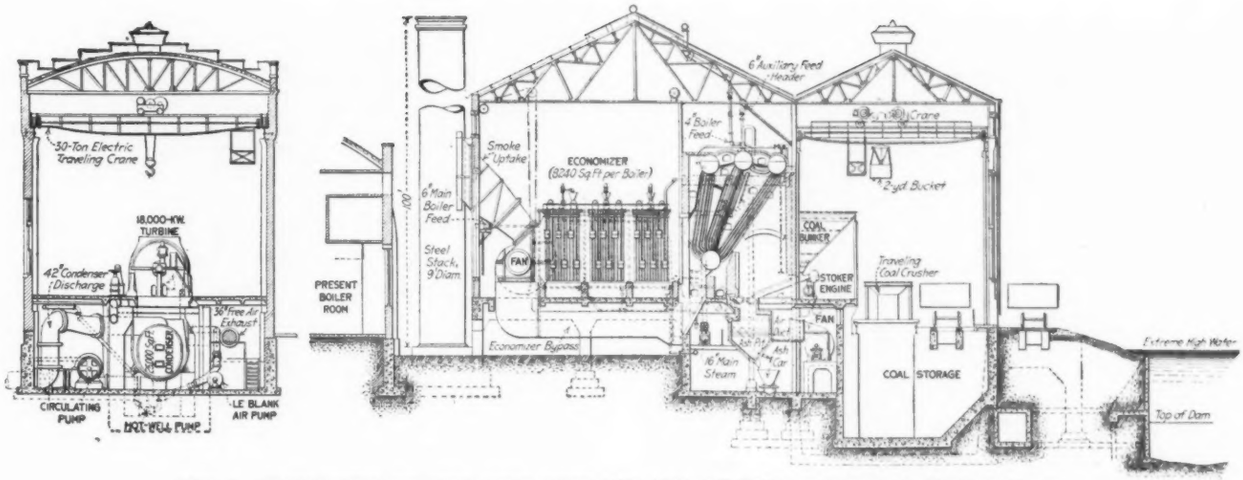


FIG. 2. ELEVATION OF THE NEW TURBINE AND OF THE BOILER-ROOM ADDITION

The boilers are designed with three banks of tube-heating surface, and the superheater is placed between the first and second bank of tubes, as shown in Fig. 2. It is designed to superheat the steam from 100 to 120 deg. according to the capacity at which the boiler is operating. Each is so designed that it is possible to increase the number of tubes at a later date if desirable and in this way obtain a higher superheat.

Each boiler furnace is equipped with a 14-retort underfeed stoker. The total grate surface is 253 sq.ft., and the entire grate area represents active surface. This gives a ratio of 54.2 sq.ft. of boiler-heating surface per square foot of grate area. The coal displaced by each piston movement is 18 lb., and each stoker of 14 retorts can handle a maximum of 16,000 lb. per hr.

The coal used has approximately the following analysis: Fixed carbon, 56 per cent.; volatile matter, 30 per cent.; ash, 10 per cent.; moisture, 3 per cent.; sulphur 1.5 per cent.; B.t.u., 13,500.

The stokers are driven by three vertical engines, each capable of driving two of the three stokers. There is a line shaft below the floor and also an arrangement of countershafts by which any stoker can be driven from any one engine, Fig. 3.

The air to the furnace is supplied by forced-draft double-inlet turbo-type fans placed in the basement on concrete piers. Each fan has a capacity of 60 000 cu.ft.

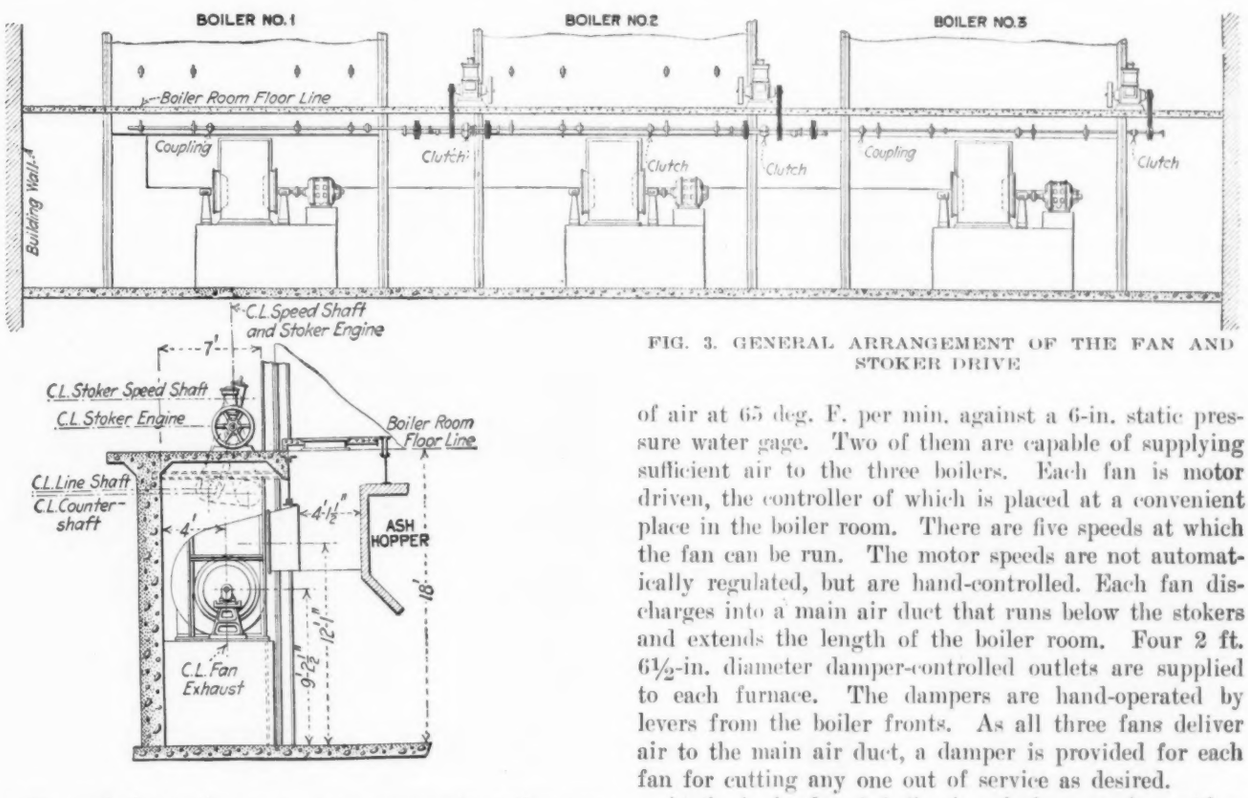


FIG. 3. GENERAL ARRANGEMENT OF THE FAN AND STOKER DRIVE

of air at 65 deg. F. per min. against a 6-in. static pressure water gage. Two of them are capable of supplying sufficient air to the three boilers. Each fan is motor driven, the controller of which is placed at a convenient place in the boiler room. There are five speeds at which the fan can be run. The motor speeds are not automatically regulated, but are hand-controlled. Each fan discharges into a main air duct that runs below the stokers and extends the length of the boiler room. Four 2 ft. 6½-in. diameter damper-controlled outlets are supplied to each furnace. The dampers are hand-operated by levers from the boiler fronts. As all three fans deliver air to the main air duct, a damper is provided for each fan for cutting any one out of service as desired.

At the back of each boiler is a fuel economizer with a total of 8,240 sq.ft. of heating surface, each unit consisting of six banks of ninety-six 12-ft. tubes. The tubes and headers are of cast iron. The economizer sections were tested to a pressure of 600 lb. in the shop prior to erection, and the whole economizer, to a pressure of 450 lb. after erection at the plant. Each economizer is ar-

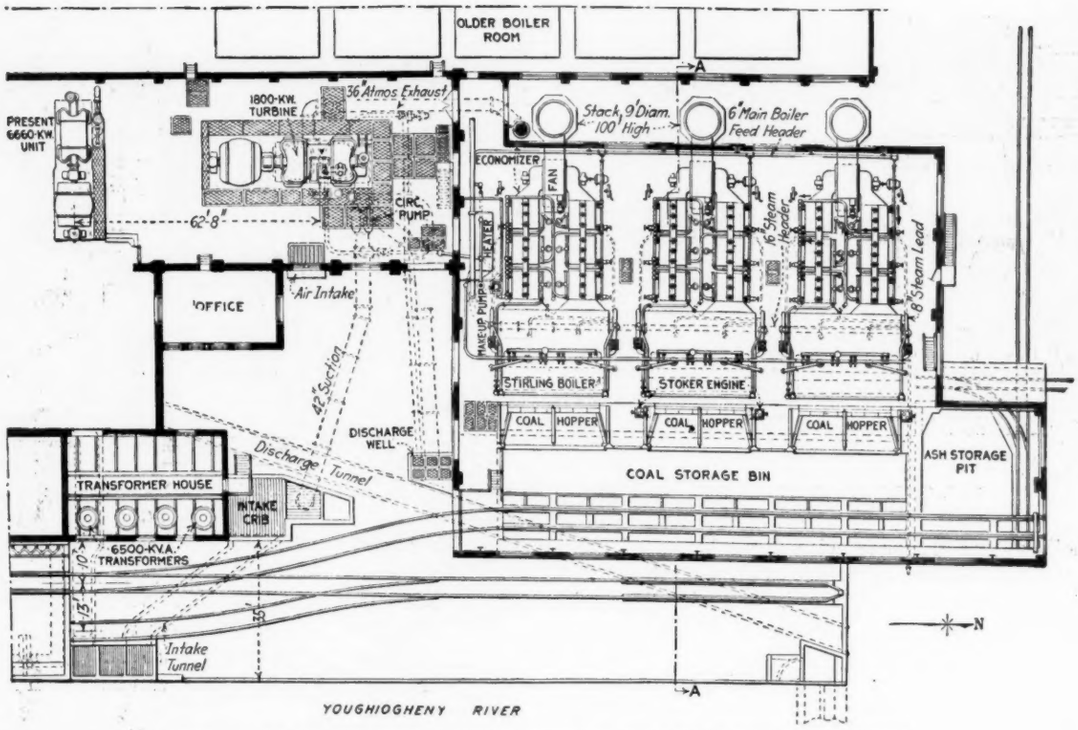


FIG. 4. PLAN OF THE CONNELLSVILLE BOILER-ROOM EXTENSION

ranged in two divisions and three banks of tubes in each division. The circulating pipes are so connected that the circulation is in a vertical direction in all the economizer tubes. The arrangement of the circulating pipes is shown in the plan view, Fig. 4.

A new idea has been incorporated in the design of the economizer, in that flanged elbows are used between the water headers and the tube headers, as shown at *A* and *B* in Fig. 5 (an end view of the outfit) and also in Fig. 6. In case a tube fractures it is necessary only to remove the elbow and secure a blank to the flange of the damaged tube header at the inlet and outlet. The economizer can then be operated without crippling the boiler until opportunity is afforded for a permanent repair.

At the outlet end of the economizer there is a double-inlet induced draft fan of the high-efficiency radial-flow type. The gases are discharged from the fan into a stack 8 ft. diameter and 90 ft. high. The fan is capable of handling 3,333 lb. of gas per minute at a temperature of 395 deg. F., and creating a draft of 2.1 in. at the fan inlet. Each fan is driven by a constant speed induction motor. Fig. 7 shows the motor, fan housing and uptake to the stack. The economizer rests on a brick foundation that also serves as a soot chamber, the soot being removed through cast-iron clean-out doors. The economizer is incased by sectional covering insulated with nonpareil insulation 4 in. thick.

The tube scrapers for each double economizer are motor-operated, speed reduction being obtained by using worm-drive gearing. The motor is placed between and at the outer end of the economizers and is so located that the worm-gear shaft is extended in both directions, from which the scrapers receive their motion.

A 5x4-ft. bypass for the furnace gases is provided under each division of the economizer as shown in Fig. 5. The passage of the gases is controlled by sliding dampers, at the inlet and outlet of each division of each economizer. The dampers are counterbalanced, the arrangement being shown in Figs. 5 and 8. The weights and dampers are suspended by a wire rope that winds around a pulley on a shaft, that is revolved by means of a hand-operated winch gearing and

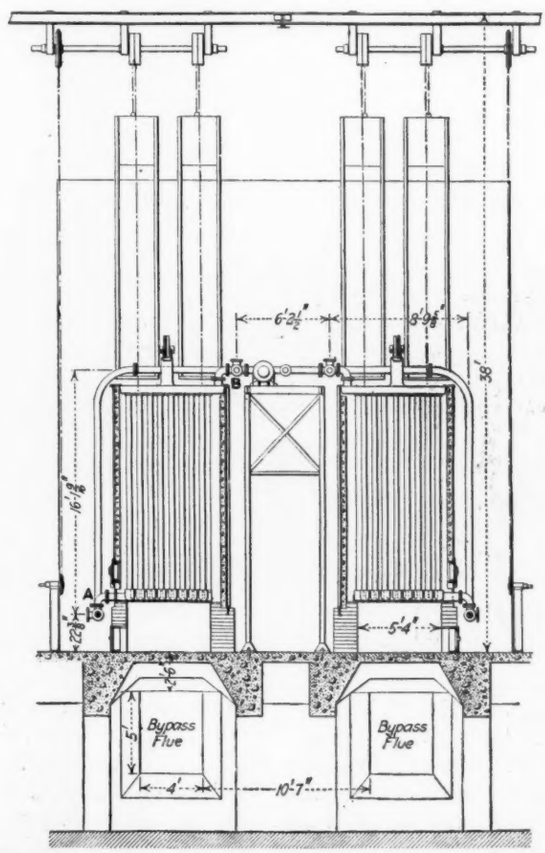


FIG. 5. END ELEVATION OF AN ECONOMIZER

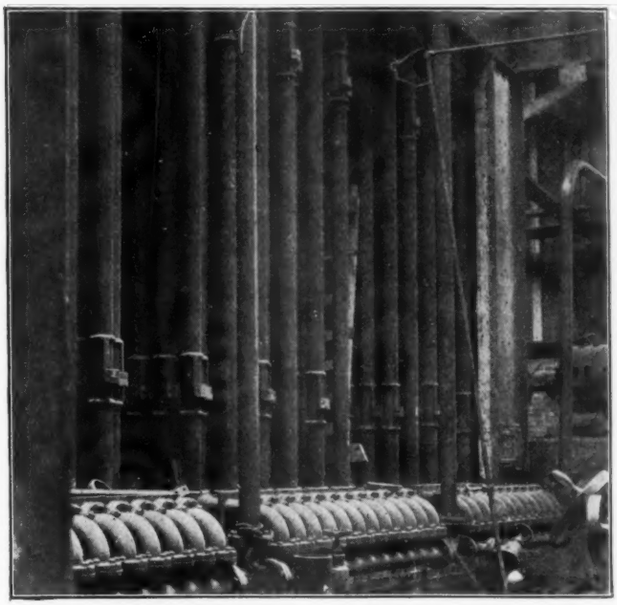


FIG. 6. SHOWING THE FLANGED ELBOWS ON ECONOMIZER

sprocket chain from the floor. The arrangement is such that when the dampers are in a position to admit gas to the economizer, they close the gas passage to the bypass and vice versa. The action of the outlet dampers is practically the same. The dampers are made with a cast-iron frame to which a sheet-steel plate is riveted in order to prevent warping.

The feed water is handled by two turbine-driven three-stage double-suction impeller, turbine centrifugal pumps, each with a capacity of 425 lb. of water per min. at 2,900 r.p.m. These are at one end of the boiler rooms. Above them on a steel platform is an open heater that uses the exhaust steam from the feed-pump turbines and from the stoker engines. It has a capacity of 250,000 lb. of water per hour at from 70 to 140 deg. An overflow pipe leads from the heater to a concrete hot-water tank, and from there the water is taken by the makeup pump to the heater. The blowoff from the economizers goes to the hot-water tank and is used again for boiler feed.

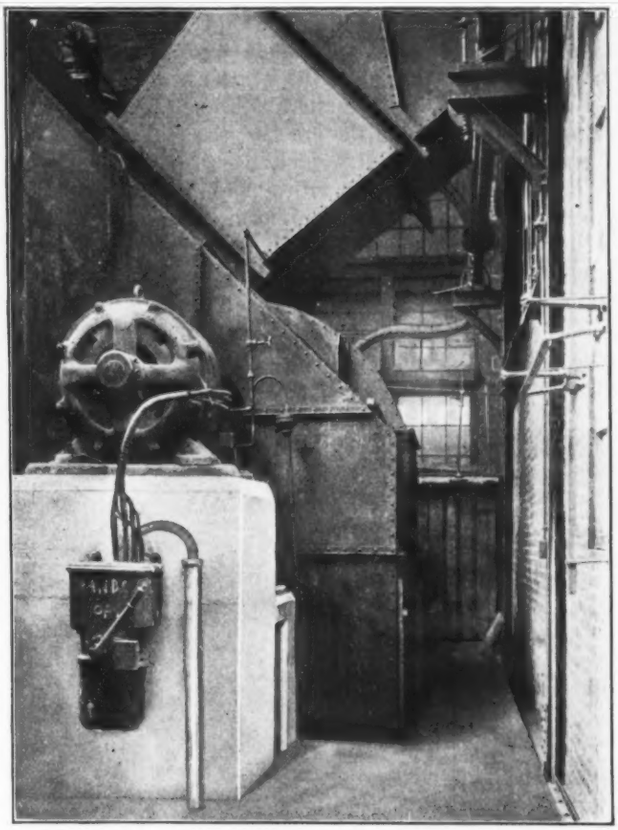


FIG. 7. INDUCED-DRAFT FAN EQUIPMENT

Coal is handled by a traveling crane with 2 cu.yd. grab-bucket (Figs. 2 and 9) from railroad cars that are switched direct into the boiler room from the yard track. The coal can be delivered to the stoker hoppers, Fig. 1, each of which has a capacity of 55 tons, or it can be unloaded into a 2,100-ton coal pit that extends the length of the boiler room in front of the boilers. If run-of-mine coal is received, it is unloaded by the crane into a four-roll crusher having a capacity of 100 tons per hour. The hopper of the crusher is large enough, about 15 tons capacity, to facilitate the rapid handling of

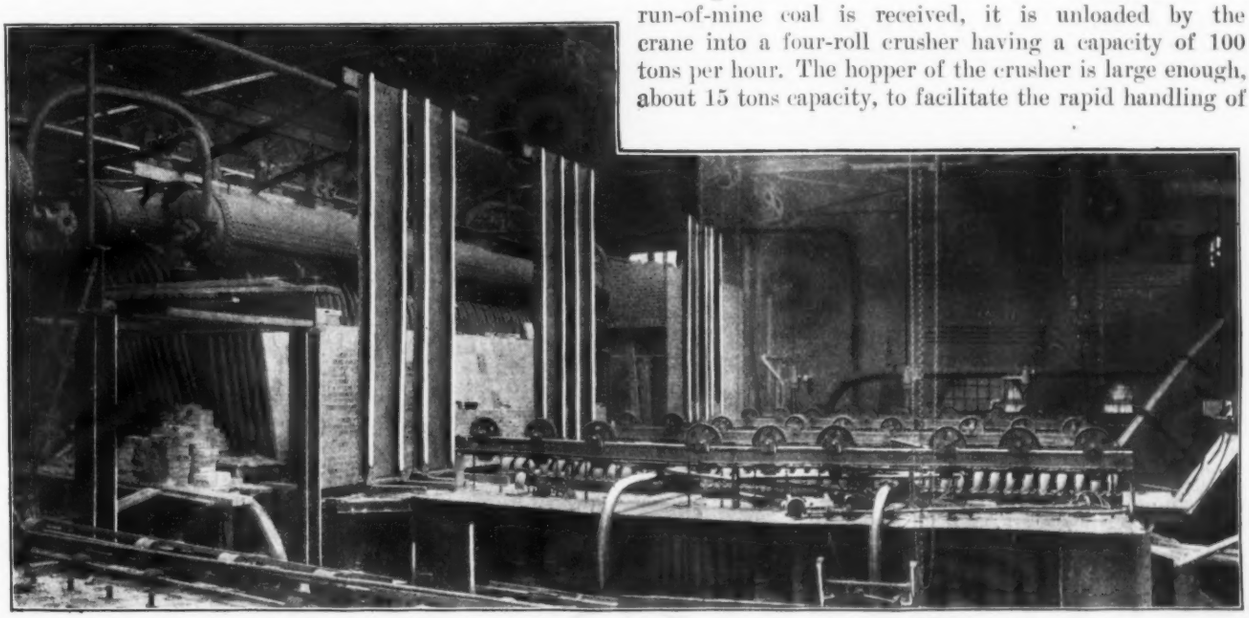


FIG. 8. VIEW ON TOP OF ECONOMIZER, SHOWING INLET-DAMPER CONTROL AND SCRAPER GEARING

coal from the crane bucket. The upper rolls are 24 in. diameter and 26 in. long. The lower rolls are corrugated and adjusted so that about 85 per cent. of the coal will pass through a 1-in. ring and all through a 1¼-in. ring. After passing through the crusher, the coal is hoisted to the stoker hoppers by the crane.

The crane is equipped with a 25-hp. lifting motor, a 35-hp. motor for opening and closing the bucket, a 5-hp. trolley motor and a 25-hp. bridge motor—all 220 volt three-phase 60 cycle. The speed of the hoist is 150 ft. per min.

The crane is equipped with a recording and weighing scale. Each time the crane bucket is hoisted loaded, the operator can weigh the coal in the bucket. The weights

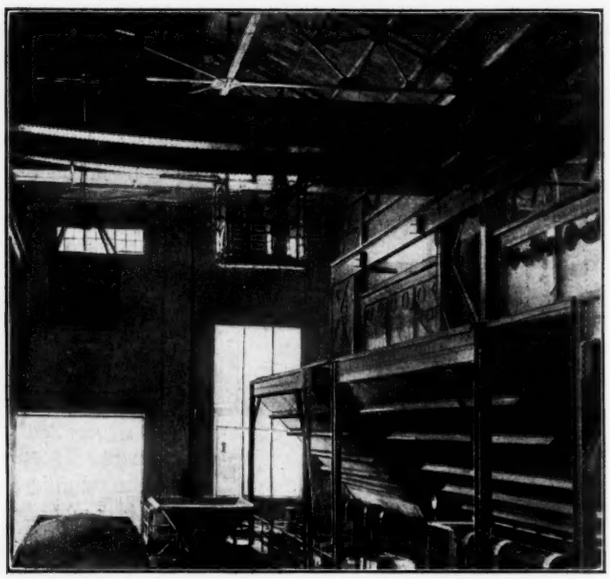


FIG. 9. COAL-HANDLING CRANE AND CRUSHER

are recorded by stamp impressions on cards which can be made in duplicate (Fig. 10). By this means the accurate weight of all coal delivered to the stokers is recorded. The ticket is self-explanatory.

Ashes from the ashpits are loaded into cars, hauled by an electric locomotive running on an industrial track (Fig. 11), and are dumped into an ash-storage pit (Fig. 4), holding approximately 320 tons. From the storage

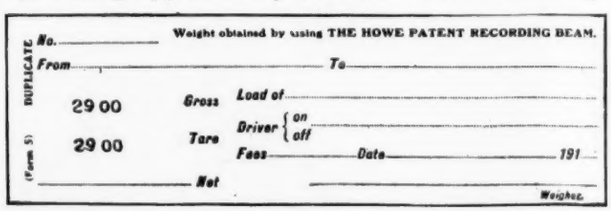


FIG. 10. CRANE COAL-WEIGHT TICKET

pit the ashes are hoisted by the same bucket-crane that handles the coal and are loaded into empty coal cars and hauled away.

The three boilers will supply steam to an 18,000-kw. maximum rating, double-flow turbine generator, which is erected in the north end of the old turbine room. The three-phase 60-cycle energy is delivered to the switches at 2,700 volts. The generator is cooled with washed air that is delivered to it at a rate of 60,000 cu.ft. per min. by a motor-driven fan. The air passes through the washer at a velocity of 700 ft. per min. Over 95 per cent. of the dirt in the air is removed in passing through the washer.

Directly below the turbine is a surface condenser having 25,000 sq.ft. cooling surface or 1.34 sq.ft. per kw.

A 22-in. motor-driven air pump of the Leblanc type is used. The motor-driven circulation pump has a capacity

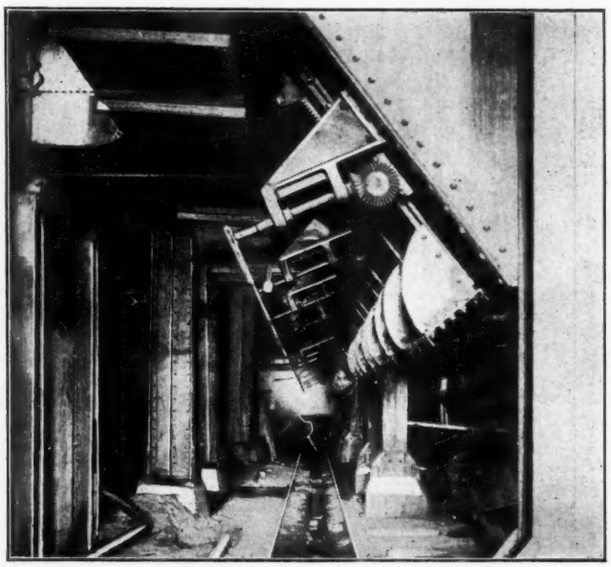


FIG. 11. ASH TUNNEL AND ASH-CAR TRACK

of 30,000 gal. of water per minute. The condensate pump is also motor driven.

Ground was broken for the addition to the boiler-house addition Sept. 23, 1915, and the first boiler was under steam Mar. 24, 1916. At the time the photographs were taken but one boiler was under pressure, the other two being in course of erection.

The erection and design of the plant were carried out by Sargent & Lundy, of Chicago.

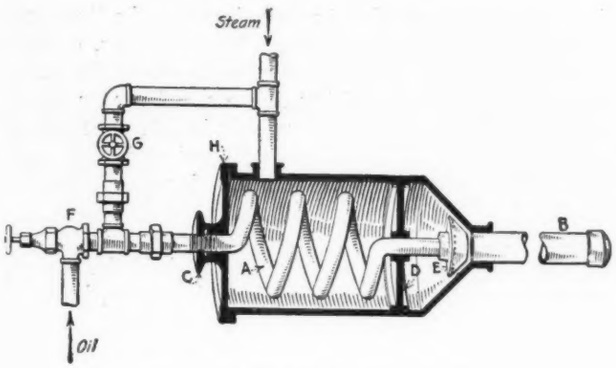
Long Fuel-Oil Burner

The following is a description of the construction and characteristics of a new fuel-oil burner as developed by G. C. Long, of Lakeland, Fla., and manufactured and sold by the Tate-Jones Co., Inc., Pittsburgh, Penn. It was designed with the view of eliminating all free oil from the furnace and to give continuous operation, wide range of capacity and as near ideal vaporization and combustion of fuel oil as possible. The flame is said to show no free oil, and at all times to be practically incandescent. It may be directed in any direction, depending on the slot in the tip within the furnace. There are no complicated parts and but two valves for oil and steam control. The burners range in capacity from 75 to 400 hp.

Referring to the illustration, the metal body contains a coil *A* and the pipe *B* conveys the steam-oil gas to the furnace. The wheel *C* is for adjusting the coil for different capacities and the coil is heated by the steam entering at the top of the body and passing around the coil. A guide *D* maintains the coil centrally in the body and *E* is a head on the coil containing slots through which the oil enters the body after being heated. A valve *F* controls the oil supply, and *G* is a valve to admit live steam to the coil for cleaning purposes.

The principle of operation is as follows: Fuel oil is admitted into the coil *A* through the valve *F*, the coil being heated to a high temperature by the steam admitted at the top of the burner and controlled by a valve not shown. The oil in passing through the coil is heated to a point at or above the flash and is ejected through a series of holes in the head *E*, where it combines with the steam that flows in a different plane at high velocity, thoroughly atomizing and vaporizing the hot oil from the coil. The vapor then flows at a reduced velocity along the pipe *B* and finally emerges into the furnace through the tip shown.

This burner is suitable for use with saturated or superheated steam and may be used as a straight-way or back-



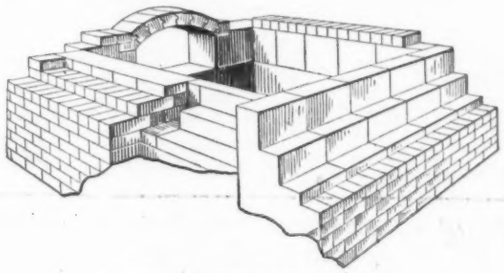
OIL-FUEL BURNER AND PIPING

shot burner. When used as a back-shot burner, the body is placed either at the front or the rear of the boiler and the mixture is carried by the pipe to the tip. For straight-way use the body is placed at the front of the boiler and the tip connected to the body by 23 or more inches of pipe. For changing the capacity and for superheated or saturated steam the adjustment is made by moving the head of the oil coil closer to or farther from the end of the body by turning the handwheel *C*, which adjustment, once made, is all that is necessary for continuous operation. For internal inspection of the body and oil coil, the faceplate is removable by disconnecting the oil feed line and steam bypass. This after removing the capscrews that hold the faceplate to the body, allows the coil to be withdrawn for inspection. The body, coiled pipe and tip need never be taken from the boiler furnace, as the tip within the furnace is not subject to carbonization because the oil is completely vaporized before leaving the tip. Small particles of carbon and tar precipitated by the vaporizing oil are carried out of the tip by the steam and oil gas, resulting in a clean body and pipe after months of service.

The Process of Galvanizing Pipe is described in the National Tube Company's "Book of Standards" as follows: The finished pipe shall be clean, freed from scale by pickling in warm dilute sulphuric acid, washed in a bath of water, immersed in an alkaline or neutral bath, then dried and immersed in molten zinc, being allowed to remain in the bath until it acquires the temperature of the zinc. No wiping or scraping device shall be used which will render the zinc coating thin.

Gates Boiler-Furnace Brick

The ordinary firebrick lining in boiler furnaces has to be renewed from one to three times yearly. This is because the first two or three courses of firebrick are fused by the high temperature of the furnace, falling to pieces and loosening up the other courses. The new type of fire-block construction designed by John W. Gates, 382



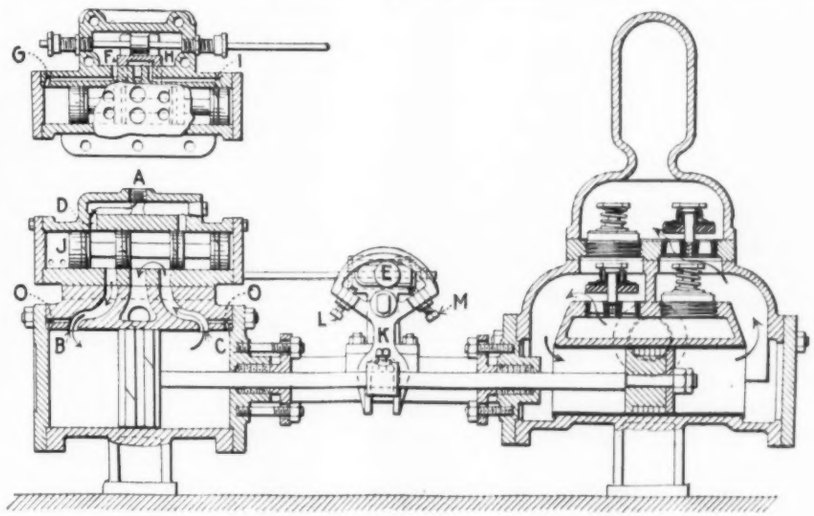
GATES FURNACE LINING

St. James St., Montreal, Canada, gives a smooth-surfaced and air-tight furnace and will last much longer than the ordinary firebrick. All blocks are interlocked, and the arch is also interlocked (see illustration) and therefore cannot easily drop. The lining can be renewed without loosening any of the other brickwork.

Piston-Valve Steam Pump

The new piston-valve steam pump recently developed by the National Steam Pump Co., Upper Sandusky, Ohio, is illustrated in section herewith.

In designing the steam end the aim has been to reduce the number of parts as much as possible, the special feature being a balanced piston valve. In the illustration the flow of steam is indicated by the arrows. Steam at boiler pressure enters the pump at *A* and alternately enters the cylinder through the ports *B* and *C*. At the beginning of the piston stroke the port *B* is covered by the steam piston and a preadmission port *O* is provided



DETAILS OF THE STEAM VALVE AND CYLINDER CONSTRUCTION

at each end of the cylinder that admits only enough steam to give the piston a start to uncover the port *B* or *C* when the piston receives full steam pressure and moves at its normal speed until it covers the port *B* or *C*, when the

remaining steam is trapped in the end of the cylinder and forms a cushion to stop the piston movement.

The valve gear is operated by the slotted lever *K* that is actuated by the piston rod and moves the guide block *E* that is secured to the valve rod.

This arrangement moves the auxiliary valve *F* in a direction opposite to that of the main piston. At the end of the stroke the auxiliary valve will be moved far enough to permit the steam to exhaust through the ports *G* and *H* and at the same time admit steam through the port *I*. This causes the piston valve *J*, when in the position opposite to that shown, to admit steam to the pump cylinder through the port *C*, the exhaust steam escapes through the port *B*. The piston valve cushions on exhaust steam in the same manner as the main piston.

The setscrews *L* and *M* are for adjusting the stroke of the pump while it is running, as will be evident from the illustration.

Distribution-Loss Factors*

By Terrell Croft

SYNOPSIS—A discussion on the losses involved in the transmission of electrical energy. The causes of these losses are pointed out and comparisons made of actual working conditions.

Distribution losses are those losses of electrical energy that occur in a transmission and distribution system between the central station feeding the community and the meter at the receiving devices on the customer's premises. These same energy losses are sometimes referred to as energy lost or unaccounted for. In every electrical-distribution plant, the total kilowatt-hours delivered to the distribution-system lines as recorded by the station totalizing watt-hour meter during a given interval of time will always be greater than the sum of the kilowatt-hours registered on all of the consumers' meters or similarly accounted for during the same interval of time. The difference between the energy thus delivered to the distributing system and that accounted for represents distribution losses, or "energy unaccounted for."

In a certain Middle-Western town of 6,000 people the central station operates a 2,200-volt primary, 110-220-volt three-wire secondary system. There are a large number of small transformers, also considerable leakage where the primary-line wires come in contact with the branches of trees. In 1914 the plant generated and delivered to the lines 348,000 kw.-hr. of energy. During the same year the energy recorded on all of the customers' watt-hour meters and otherwise accounted for was only 251,000 kw.-hr. Hence, in this instance the distribution losses were 348,000 — 251,000 = 97,000 kw.-hr.

In a town of less than 1,000 inhabitants in Iowa, where a 110-220-volt alternating-current three-wire system (without transformers) is used for distribution, the central station delivered in 1913 to the lines 25,000 kw.-hr. The customers' meters for the same interval recorded 23,000 kw.-hr. Hence, for this plant and this year, the distribution losses were, 25,000 — 23,000 = 2,000 kw.-hr.

For the year ending June 30, 1915, the Pacific Power and Light Co., which operates in Oregon and southern Washington, generated or bought 45,473,923 kw.-hr. of energy. For the same period the energy delivered to customers or otherwise accounted for was only 37,746,854 kw.-hr. Hence, the distribution losses for that year were 7,727,069 kw.-hr.

Distribution losses include all energy that is not delivered to customers or otherwise accounted for, such as: Line losses, made up of I^2R energy losses in the line conductors, mains and service wires; leakage losses, due to insufficient insulation, grounds against trees and the like; transformer losses, comprising the iron and copper losses in the transformers (these occur only in alternating-current plants); meter losses, caused by slow meters and electrical losses within the meters; stolen-energy losses, occasioned by theft of current.

The line loss—that is, the kilowatt-hours lost in the line conductors—can be readily computed if the resistance of the line and the current in it is known, because *Watts line loss* = (*line current in amp.*)^2 × (*resistance of line in ohms*). Then, if the watts loss thus obtained be multiplied by the number of hours during which the current flows and divided by 1,000 the result will be the kilowatt-hours lost in the line. Since the current in a distribution line is seldom constant, it is necessary to recognize this condition in computing line loss. The line current will vary from hour to hour and from month to month. However, the approximate loss can be readily calculated if 24-hr. load curves for four typical months of the year—say March, June, September and December—are available. The load factor of the plant has, obviously, a bearing on its line loss. The process is a trifle tedious, and a detailed description of it would be out of place here. For a complete discussion of the method, see Gear and Williams' "Electrical Central Station Distributing Systems," page 274.

The line loss may be either a large or a small proportion of the distribution loss, depending on whether the largest or the smallest feasible conductors are used for the distribution lines. If the designer provides excessively large conductors, the line loss will then be very small. However, as plants are usually designed, the line loss is relatively small.

The leakage loss will be determined wholly by the thoroughness with which the line was originally constructed and by the effectiveness of its maintenance. If the conductors are supported on insulators of proper design and material and are held away from tree branches on insulators and if the trees through which the line passes are well trimmed, the leakage loss will be very small. There is no practicable method of computing these losses. They can be determined by test, but this is usually impracticable, because it involves the simultaneous opening of every consumer's service switch.

Transformer losses are subdivided into copper losses and core losses. The copper loss of each transformer varies with its load; with no load on a transformer it is very small. The iron losses are practically constant as

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long as normal line voltage is impressed across the primary terminals of the transformer, whether the secondary is loaded or not. The transformer losses are likely to be a large proportion of the total distribution loss, particularly where the load is mostly lighting and the plant operates 24 hr. per day, because the core losses are going on every hour that the transformer is connected to the line. Small underloaded transformers are a source of excessive loss; hence the capacities of transformers should be carefully determined so that, in general, a few large, fully loaded transformers will be used rather than many small, underloaded ones.

The transformer losses of a system may be computed approximately if the ratings of the different transformers, their efficiencies and the loads and duration thereof which are imposed on them are known.

Meter Losses and Stolen Energy

Meter losses are relatively small, although they may in the aggregate be, contrary to the generally accepted opinion, greater than the line losses. The power lost in the shunt or voltage coils of watt-hour meters will probably range between one and four watts. Meters of the older designs appear to have the greater losses. The power loss in the voltage coil of an average modern meter is in the neighborhood of 2.4 watts. The power lost in the current coil is very small, almost negligible even when it does occur, and it occurs only when current to serve a load is passing through the meter. But the loss in the voltage coil occurs continuously as long as voltage is impressed on the meter. The total aggregate energy loss thus involved may be relatively considerable, as indicated by the following example:

If the power loss in the voltage coil of a watt-hour meter is 1.5 watts and the meter is connected ot the lines of a plant giving 24-hr. service, what will be the energy loss in this coil in a year? There are 8,760 hr. in a year. Hence the energy loss would be 1.5 *watts* $\times$ 8,760 *hr.* = 13,140 *watt-hr.* = 13.1 *kw.-hr.*

Stolen energy loss is obviously difficult of determination. Whether or not it assumes material values depends largely on the policy and vigilance of the concern that is giving service. However, in any case it is likely to be but a small proportion of the total distribution loss.

A specific numerical illustration of distribution losses and the segregation thereof is given in Table 1 and was computed from actual operating data for the year 1913, of a central-station plant in a city of about 6,500 inhabitants. The 60-cycle two-phase distribution system under consideration comprises 2,400 volts primary and a 110-220-volt three-wire secondary. While the distribution losses indicated are greater than they would be in a well-designed distribution plant, there are probably many small central-station systems that cannot show a much better performance.

During the year 1913, 250,000 kw.-hr. of electrical energy was delivered to the customers of this central station. During this same interval, there was supplied to the distribution system 350,000 kw.-hr. Hence the distribution loss was, 350,000 — 250,000 = 100,000 kw.-hr. It was estimated, on the basis of careful calculations, that the annual line transformer and meter losses were about as shown in the table. No specific estimates of leakage loss or stolen energy were made. These were included in the three loss items that were estimated.

The distribution plant included 60 transformers ranging in size from 2 to 40 kv.-a., all but four being under 20 kv.-a. in capacity. Probably many of these transformers were of old and inefficient types. There were 973 watt-hour meters in the installation, ranging in capacity from 200 amp. to 5 amp.; all but 11 were of 50-amp. or less capacity and 704 were 5-amp. instruments.

A consideration of these data will emphasize the importance of using only high-efficiency transformers if the distribution losses are to be maintained at a minimum, because in this example the transformer loss was much the greatest of all the distribution losses. The meter loss was greater by 5,000 kw.-hr. than the line loss. These data show that the usually accepted notion that the line loss is always the greatest of the distribution losses may not, by any means, always be correct.

TABLE 1. LOSSES IN DISTRIBUTION

Item	Annual Kilowatt-Hours	In per Cent. of Energy Generated	In per Cent. of Energy Delivered
Line loss	10,000	2.8	4.0
Transformer loss	75,000	21.4	30.0
Meter loss	15,000	4.3	6.0
Total distribution loss	100,000	28.5	40.0
Supplied to system	350,000	100.0	140.0
Delivered to consumers and accounted for	250,000	71.5	100.0

The distribution-loss factor is that value, relating to some particular system, expressed as a percentage, which, if the *energy delivered and accounted for* be multiplied by it, will give the *energy lost in distribution.* It is the per cent. of energy "sold" which is lost and unaccounted for in distribution. Therefore,

$$\textit{Distribution-loss factor} = \frac{\textit{kw.-hr. distribution loss}}{\textit{kw.-hr. delivered and accounted for}} \quad (1)$$

$$\textit{Kw.-hr. dist. loss} = (\textit{dist.-loss factor}) \times (\textit{kw.-hr. del. and acc. for}) \quad (2)$$

$$\textit{Kw.-hr. del. and acc. for} = \frac{\textit{kw.-hr. dist. loss}}{\textit{dist.-loss factor}} \quad (3)$$

Notice that the kilowatt-hours delivered and the kilowatt-hours lost must both be measured over the same interval of time—preferably over an extended period such as six months or a year.

In the case of the foregoing system the energy supplied to the system in 1913 was 350,000 kw.-hr. During the same year the energy delivered to consumers was 250,000 kw.-hr. What was the distribution-loss factor for this plant during this period? From equation (1)

$$\textit{Dist.-loss factor} = \frac{\textit{kw.-hr. dist. loss}}{\textit{kw.-hr. del. and acc. for}} = \frac{100{,}000}{250{,}000}$$

$$= 0.40 = 40 \textit{ per cent.}$$

Hence, the distribution-loss factor for this plant for the year 1913 was 40 per cent.

In a certain small town of 500 inhabitants it was estimated that the total energy sold would be 16,370 kw.-hr. annually. If it be decided that the town will be served by an alternating-current 2,400-volt primary 110-volt secondary system and that a loss factor of 20 per cent. be assumed, how many kilowatt-hours will have to be generated annually? From equation (2)

Kw.-hr. dist. loss = (*dist.-loss factor*) $\times$ (*kw.-hr. del. and acc. for*) = 0.20 $\times$ 16,370 = 3,274 *kw.-hr.*

That is, the annual distribution loss would be 3,274 kw.-hr. Then there would have to be generated, 16,370 + 3,274 = 19,644 kw.-hr. Or, a more direct solution is, 16,370 $\times$ 1.20 = 19,644 kw.-hr.

In a certain central-station plant there is 789,600 kw.-hr. supplied to the distribution system annually. If the distribution-loss factor for this system is assumed to be 25 per cent., how much energy is delivered to customers and otherwise accounted for annually? From the preceding discussion it follows that

$$Kw.\text{-}hr.\ del.\ and\ acc.\ for = \frac{kw.\text{-}hr.\ supplied\ to\ dist.\ system}{100 + dist.\text{-}loss\ factor}$$

$$= \frac{789{,}600}{1.25} = 632{,}000\ kw.\text{-}hr.$$

Hence 632,000 kw.-hr. of energy would annually be delivered to the customers of this system.

Probable distribution-loss factors—that is, factors that will probably apply for certain different conditions of service—are given in Table 2. These are based on data from a number of cases encountered in actual practice and are believed to be representative of the "Probable Fair Average Value"; value that may be used in making estimates. It should be understood, as previously suggested, that the distribution-loss factor for any certain installation will be determined wholly by the characteristics of that system.

TABLE 2. APPROXIMATE DISTRIBUTION-LOSS FACTORS

Kind of Plant and General Conditions	Distribution-Loss Factors Probable Range	Probable Fair Average Value
Alternating Current—		
Without transformers and well designed; 110-220 three-wire or 440 volts	10 to 20	15
Well-designed system; 2,200, 2,400 or 6,600 volts primary and 220 or 440 volts secondary; largely power load of reasonably high-load factor	15 to 25	20
Well-designed system; 2,200 or 2,400 volts primary and 110-220 volts three-wire secondary; general lighting and power load	20 to 30	25
Poorly designed system; large number of small transformers; 2,200 or 2,400 volts primary and 110-220 volts secondary; general lighting and power load	25 to 45	35
Direct Current—		
Well-designed; lighting and power load	5 to 20	10
Poorly designed; lighting and power load	10 to 25	15

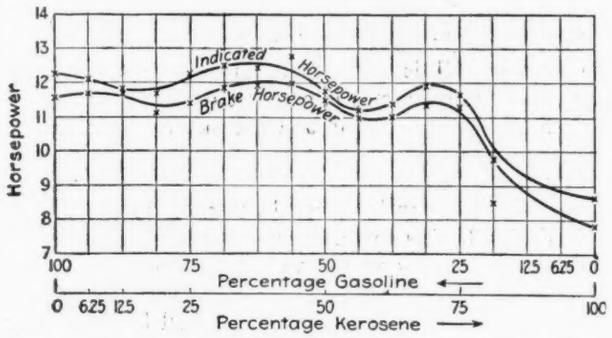
For a central station where a residence-lighting load predominates and there is little power load, the distribution-loss factor will be much higher than where the power load is predominant. The reason for this is that with the residence-lighting load the transformer loss will be proportionally large. It is also true that distribution loss will usually be greater, relatively, for a residence district where the consumers are widely scattered than for a district where the loading is dense, for the reason that a few large, well-loaded, efficient transformers can be used for serving the dense load, while the scattered load will probably be fed by many small, underloaded, less-efficient transformers. One case at least has been recorded where, largely the result of underloaded transformers, the distribution loss in a small plant amounted to 50 per cent. of the energy sold.

A line-loss factor is a value representing the ratio of the actual I^2R energy loss in a feeder or other line-circuit component during a year to the I^2R energy loss that would have occurred in that feeder, or other circuit component, if it had carried continuously the maximum load ever imposed on it during the entire year. This factor is sometimes indefinitely referred to as merely "loss factor." It is somewhat similar in derivation to load factor, which is the ratio of the average load imposed on a station or system to the maximum load imposed on it. Ordinarily, because the line current is changing constantly, it is impossible to obtain the actual I^2R energy loss in a circuit, hence in computing line-loss factor in practice, an estimated approximate value is determined in accordance with the process referred to in a preceding paragraph.

Power factor affects distribution loss to some extent. If the power factor of the receiving apparatus is less than 100 per cent., a proportionately larger current will be necessary in the line conductors to produce a given amount of power at the consumers' premises than if the power factor is unity. Consequently, with low power factor the line loss and the transformer copper loss may be materially increased.

Tests with Gasoline-Kerosene Mixtures

A series of tests to determine the relative power output of a 15-hp. four-cycle gasoline engine, when supplied with different mixtures of gasoline and kerosene, were recently made by Messrs. McGinnis, Hutchinson and Udy, as a graduating thesis at the Virginia Polytechnic Institute.



CURVES SHOWING VARIATION OF HORSEPOWER WITH MIXTURE COMPOSITION

The engine was first started on gasoline and allowed to run until it became thoroughly warm, then was quickly switched over to the mixture of gasoline and kerosene. As will be seen from the curves, the horsepower output gradually increased as the percentage of kerosene was increased, up to about 50 per cent., at which point it began to fall off rapidly. At about 60 per cent. of kerosene the engine began to pound, this pounding being more and more marked as the percentage of kerosene increased, until near 90 per cent., when it became necessary to retard the spark with consequent reduced pounding, but also a decrease in power.

Although the exhaust was led to a sewer, there was a noticeable odor from the burning of the kerosene when its proportion reached about 40 per cent. This odor became more obnoxious as the mixture grew stronger in kerosene, and at about 80 per cent. the atmosphere of the room had a blue appearance, due to the presence of the partly burned products leaking from the flanges of the exhaust pipe and escaping past the piston.

Greatest Pressure on Guides—At what point of the stroke of an engine is the pressure greatest on the guides? J. R.

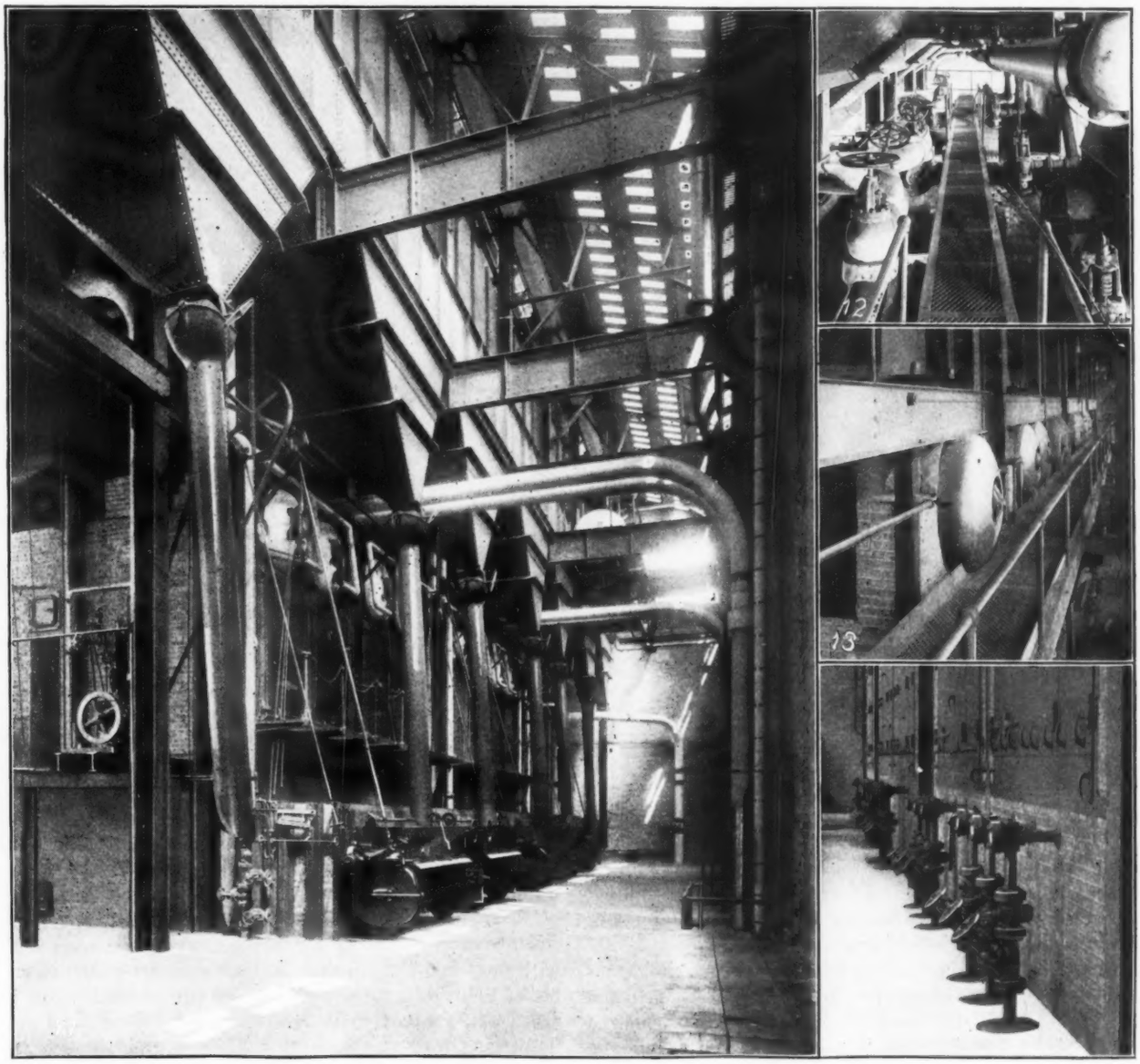
The greatest pressure on the guides occurs at that point of the stroke when the greatest product is obtained by multiplying together the pressure on the piston and the distance the crank is to one side of the center line of the cylinder.

Hot-Water Heating at Crane Plant, Chicago--II

SYNOPSIS—Brief description of boiler, refrigeration and compressor plants serving the works. Coal consumption and results.

The power-plant building containing the boilers, air compressors, refrigerating system and the heaters and pumps has been designed to economize space to minimize the overhead expense. The various pumps are placed under the firing aisle of the boiler room, the circulating pumps of the refrigerating system under the generator, and in other ways the installation has been made as compact as possible. At the same time all equipment is accessible, and particular pains have been taken to locate the various valves on pump lines, etc., within easy reach of the operator. The valves for each pump or each unit are grouped so that a change from one line to the other may be readily effected.

At present the boiler room contains four water-tube boilers each having 5,080 sq.ft. of heating surface, and two more of the same size are being installed. Chain-grate stokers with an active grate area of 102 sq.ft., bearing a ratio to the heating surface of 1 to 50, serve the boilers. There are ash and fine-coal hoppers under each stoker, and industrial cars remove the former and return the green coal to storage. The boilers are designed for 200 lb. pressure, although the working pressure is 150 lb. gage, and arrangements have been made



FIGS. 11 TO 14. SOME GENERAL VIEWS ABOUT THE BOILER ROOM
Fig. 11—Boiler-room firing aisle. Fig. 12—Walk above boilers. Fig. 13—Walk at rear of drums. Fig. 14—Angle blowoff valves and cocks

for the future installation of superheaters. The furnace has an extension of 4 ft. 6 in., and the headroom under the bottom of the front header is 10 ft. The tubes have been vertically baffled for three passes, the gases passing around the rear of the drums to the uptake.

To supply steam at 350 lb. for testing purposes in the shops, one boiler containing 2,500 sq.ft. of heating surface has been installed, and there is space for a second of the same size. There is also a separately fired superheater of the three-pass type, which will superheat the steam 564 deg., or to a final temperature of 1,000 deg. F.

These boilers and the superheater are served by a steel breeching lined with 1½-in. vitrified asbestos board and ½ in. of asbestos finish. Through an opening containing 134 sq.ft. the gases are discharged into a radial brick stack having an inside diameter of 12½ ft. and rising 225 ft. above the boiler-room floor line.

Coal is taken from railway cars by a two-drum monorail electric hoist equipped with a 1½-yd. bucket. The hoisting speed of this crane is 100 ft. per min. and the travel speed, 500 ft. From railway car to bunkers the crane has a capacity to transfer 50 tons of coal per hour. Individual 60-ton steel bunkers serve each boiler. There is no weighing hopper or traveling scale, so the coal is chuted directly to the stoker hoppers.

Some of the features of the boiler room are a walk along the rear of the boilers to facilitate taking off manhole caps and another walk over the boilers from which the safety pops and the boiler check valves can be easily reached. The blowoff valves are well above the floor and readily accessible. The blowoff tank, located in the basement, is equipped with a coil through which the makeup water passes and absorbs some heat that would otherwise be wasted. This tank, which is designed to withstand 250 lb. pressure, also has the usual vent and overflow.

Steam is conveyed from each battery of two boilers by 6-in. pipes leading to a steel manifold in the basement. Each unit is connected to the 8-in. air-compressor header, the 10-in. main supplying live steam to the heating system and the shops and to a 5-in. auxiliary steam header supplying the equipment in the pumproom. Each manifold has a drip pocket which is the lowest point in the steam-distributing system. This is served by a tilting trap which returns the condensation to the feedwater heaters.

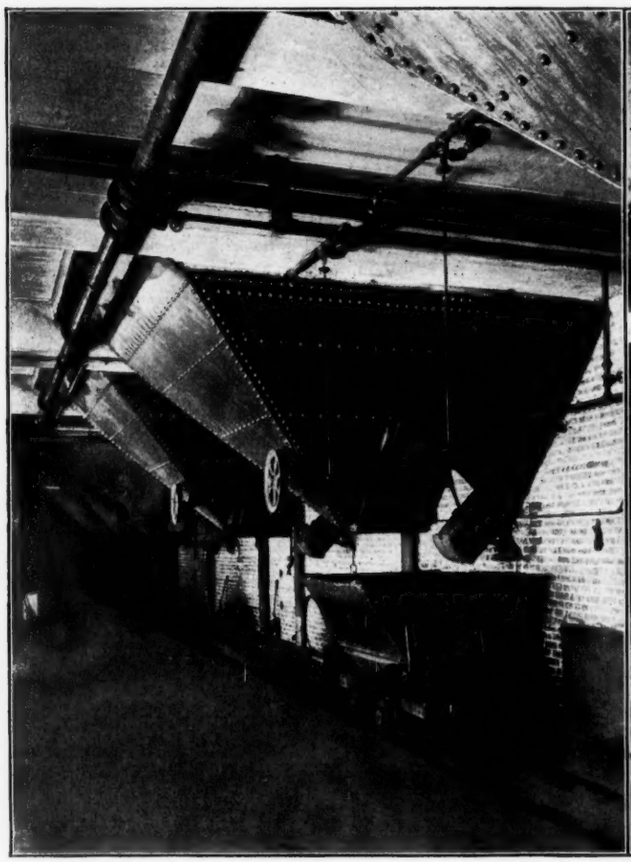


FIG. 15. ASH AND FINE-COAL HOPPERS

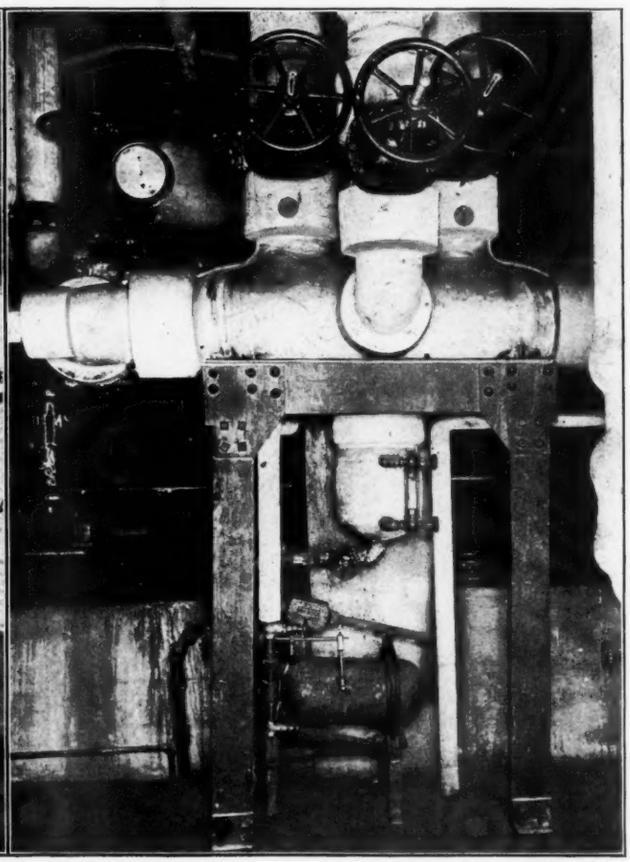


FIG. 16. STEEL MANIFOLD HEADER IN BASEMENT

To supply compressed air at 90 lb. pressure for miscellaneous purposes in the shops, two air compressors with cross-compound steam and air cylinders, each capable of delivering 2,500 cu.ft. of free air per minute at a speed of 97 r.p.m., have been installed. Provisions are now being made for a third machine of the same type, which will deliver 4,375 cu.ft. of free air per minute. The demand for compressed air has been such that the two compressors have been operated at full capacity during the day and one at night. With the addition of new buildings more compressed air will be required, so that the services of the new compressor will be needed.

Two heaters having a capacity of 5,000 gal. per hr. each, with shells 10 ft. long and 5 ft. 6 in. diameter, supply hot water for domestic service. Each contains

490 sq.ft. of heating surface in 1½-in. No. 15 gage brass tubes 46 in. long. A diaphragm valve under the Powers system of controls admits enough exhaust steam to keep the water at the proper temperature, and in addition, reducing valves admit live steam if necessary.

A 30-ton absorption refrigerating system cools the drinking water for the 115 fountains now installed; there will be about 225 fountains when the works are completed. The generator is built for 50 tons' capacity, and provisions have been made for adding two more condenser coils. The condenser and weak-liquor cooler are of double-pipe construction, receiving cooling water from a 4,000,000-gal. pond through 8-in. suction and discharge lines which are 1,200 ft. long and capable of supplying 800 gal. per min. This line also serves the air-compressor jackets. Instead of passing the water through the various elements of the absorption system in series, it is fed through multiple connections, the discharge from each being returned to the pond.

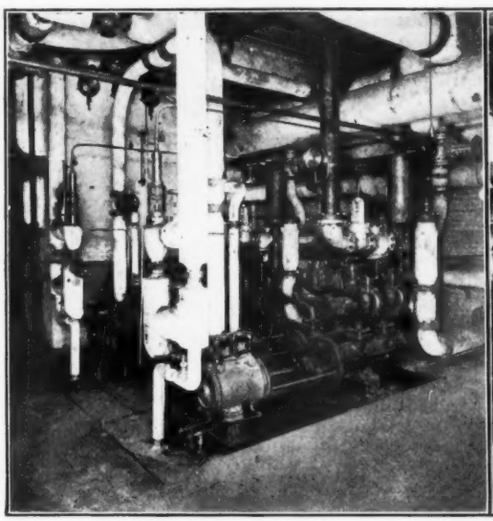


FIG. 17. HIGH-PRESSURE BOILER FEED PUMPS

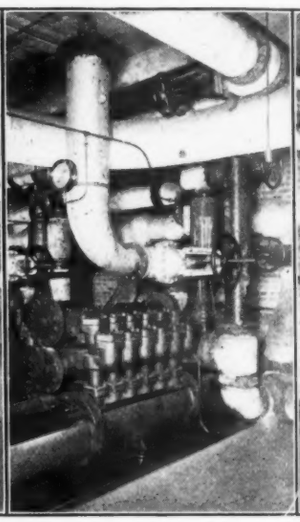


FIG. 18. MEDIUM-PRESSURE FEED PUMP

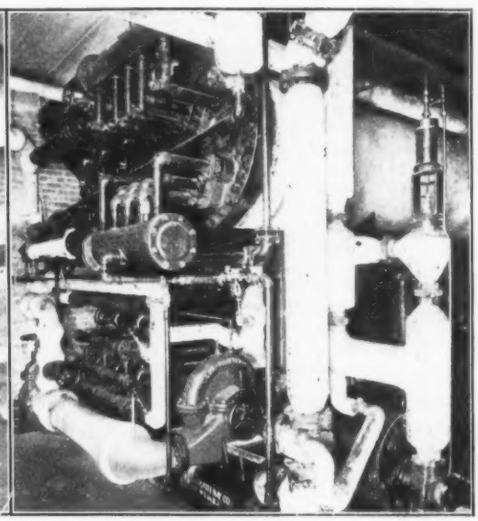


FIG. 19. ABSORPTION REFRIGERATING SYSTEM

From the preceding some idea will have been obtained of the services required from the plant during the first few months of operation. Although it is intended to install meters so that the various services may be separated, at present this has not been done. The coal consumption, beginning with October, 1915, and ending with April of the present year, totaled 12,002 tons. In respective order the tonnage for each month was 1,054, 1,100, 1,964, 2,132, 2,349, 1,914 and 1,489.

The coal was used for all services, including the steam for testing purposes in the shops. Without meters it is impossible to apportion accurately the steam and coal for each service. As the exhaust-steam supply and the demand are practically balanced, it is evident, however, that the heating system must have operated at high economy; it has given a minimum of trouble. The back pressure on the air compressors has been 2 lb. or less, and owing to the automatic regulation of the supply, adjusted hourly according to schedule, the service has been most satisfactory.

The entire heating installation was laid out and its installation supervised by F. E. McCreary, of the Crane Co. C. Gerling, of the same company, did the drafting work.

Accounting at Pasadena Plant

By L. R. W. Allison

The Pasadena, Calif., municipal lighting plant, one of the most noted city-operated stations in the country, has adopted an effective and interesting system of accounting for all departments of operation. This system as arranged is particularly comprehensive, providing not only for the requirements of the plant, but harmonizing as well with the classification of accounts for public utilities as devised by the California Railroad Commission.

Whether or not state utility laws prevail for the regulation of this phase of power-station operation, the large or small plant in justice to itself should seek to install a proper schedule of accounting to suit the particular needs—a system that will afford at any time an immediate and thorough survey of existing conditions. An established and perfected system as now operative at the Pasadena station is especially instructive in rendering a concrete idea of the value and possibilities of a simple, accurate and adequate classification of station and outside operating accounts.

In considering the features of this accounting system it is of interest to mention that the Pasadena plant has a present generating capacity of 6,300 kw., having been recently increased to this total by the installation of a 3,300-kw. turbine. The plant equipment consists of boilers of the B. & W. and Stirling water-tube types; a cross-compound condensing engine of 2,000-hp. rating and a Parsons steam turbine of like capacity, in addition to the new unit. The system embraces a city area of about 11 sq.mi., with nearly 200 mi. of pole lines, having a total of over 9,000 service connections serving an aggregate of 8,100 consumers. The plant employs on an average from 60 to 70 men.

The efficiency of the station is attested to by the existing rates for light and power, which, as is generally known, are among the lowest in the country. The charge for lighting service is 5c. per kw.-hr.; and for power, 4c., with a sliding scale of reduction for large consumption, reaching to a point of 3c. for lighting where over 2,000 kw.-hr. are used in a month, and 1.2c. for power service of over 3,000 kw.-hr. per month. The minimum monthly

charge for light and power connections is 60c. and $1 respectively.

The accounting system as adopted by the plant is divided into two major schedules of "Construction" and "Operation." The first is subdivided under three primary heads of "Production Capital." "Distribution Capital" and "General Capital." The "Operation" schedule is arranged under four primary accounts of "Production," "Distribution," "Commercial" and "General." These classifications include every branch of operation, systematizing the accounting in direct and independent schedules, yet readily linked together for a general statement of current conditions under a head of surplus or deficit.

CONSTRUCTION

Production Capital:
1. Land devoted to electric operations
2. Power-plant buildings
3. Furnaces, boilers and accessories
   (a) Boilers
   (b) Furnaces
   (c) Piping
   (d) Auxiliaries
4. Steam and power-plant equipment
   (a) Engines and turbines
   (b) Electric generators
   (c) Switchboard and cables
   (d) Transformers and electric auxiliaries
   (e) Cooling tower
   (f) Auxiliaries

Distribution Capital:
5. Poles and fixtures
   (a) Poles
   (b) Fixtures, crossarms, etc.
6. Overhead system
   (a) Wire
   (b) Miscellaneous, insulators, etc.
7. Underground conduits
   (a) Conduits
   (b) Cables, terminals, etc.
8. Substation buildings
9. Substation equipment
10. Line transformers and devices
   (a) Transformers
   (b) Devices
11. Electric services
12. Electric meters
13. Municipal street-lighting system
   (a) Poles
   (b) Wire
   (c) Arcs
   (d) Incandescent fixtures
   (e) Underground conduits, etc.
   (f) Underground cables and devices, terminals, etc.
   (g) Miscellaneous

General Capital:
14. General structures
15. General equipment
   (a) Office
   (b) Shop
   (c) Store
   (d) Garage
16. Telephone lines

These 16 primary accounts, with auxiliary divisions, comprise the entire schedule under the head of "Construction." Each account is self-explanatory as regards the respective classification and just what is to be included thereunder. This division embraces practically every important function of construction work as found in the ordinary power station and represents a comprehensive and valuable summary for estimating funds and resources devoted to this phase of operation.

OPERATION

Production:
1. Fuel for steam
2. Steam-generator supplies
   (a) Water
   (b) Lubricants
   (c) Miscellaneous supplies
3. Steam- and electric-plant labor
4. Electric-plant supplies
5. Repairs to power-plant buildings
6. Repairs to boilers, furnaces and accessories
   (a) Boilers
   (b) Auxiliaries, including steam piping
7. Repairs to steam power-plant equipment
   (a) Engines
   (b) Generators
   (c) Switchboard and cables
   (d) Auxiliaries

Distribution:
8. Superintendence
9. Setting and removing transformers, meters and services
   (a) Transformers
   (b) Meters
   (c) Services
10. Inspecting and patroling
11. Electric-meter operations, testing and cleaning
12. Commercial incandescent-lamp renewals
   (a) Mazdas and nitrogens
   (b) Gems and carbons
13. Municipal street arc and incandescent labor
   (a) Arc labor
   (b) Incandescent labor
14. Municipal street arc and incandescent supplies
   (a) Arc supplies
   (b) Incandescent street lamps
   (c) Globes
15. General labor and supplies
16. Repairs to substation building
17. Repairs to substation equipment
18. Repairs to overhead distribution system
   (a) Poles
   (b) Fixtures
19. Repairs to underground distribution system
   (a) Conduits
   (b) Cables, etc.
20. Repairs to line transformers and devices
   (a) Transformers
   (b) Devices
21. Repairs to electric services
   (a) Fuses
   (b) Other repairs
22. Repairs to meters
23. Repairs to municipal street-lighting system
   (a) Poles
   (b) Fixtures
   (c) Conduits
   (d) Cables
   (e) Arc lamps

Commercial:
24. New business
25. Commercial-department salaries and expenses
26. Commercial-department indexing
27. Commercial-department collections

General:
28. Salaries of general officers
29. Salaries of general-office clerks
30. General-office supplies
31. Law expenses
32. Injuries and damages
33. Insurance
34. Repairs to general structures
35. Repairs to general equipment—office
36. Repairs to general equipment—shop
37. Repairs to general equipment—store
38. Repairs to general equipment—garage, including labor and materials
39. Repairs to telephone lines
40. Extraordinary repairs
41. Uncollectible bills
42. Rents received
43. Rents paid
44. Electric merchandise

These 44 primary accounts, with the necessary subdivisions, make up the entire classification under the head of "Operation." As in the case of "Construction Accounts," each schedule is self-explanatory.

A general digest of this accounting system shows its simplicity and adaptability for the service intended. It also affords a flexibility for any additions or eliminations that might be required, making it an ideal basis for accounting classifications for the power station operating along standard and recognized lines. It is a system that incorporates every essential feature of power-plant work, with a thorough and practical arrangement of schedules for figuring costs and revenues, profits or losses.

Testing Refrigerating Plants

By Julian C. Smallwood*

SYNOPSIS—Essentials in testing an ammonia compression plant and definitions of the usual quantities sought.

Tests of refrigerating machinery are in general more difficult to make and get accurate results from than tests of other classes of power plant. It is perhaps for that reason that authoritative and complete data upon such tests are comparatively meager. With the growth of the industry (which, the writer foresees, will increase vastly in the near future) the demand for accurate measurements of capacity, economy and efficiency as commercial requirements will become imperative, as it has in all types of industrial heat and energy transfers.

It is the purpose here to relate briefly the essentials of the test of a refrigerating plant. For the present, the compression ammonia system only will be considered. Although they are more or less familiar, it is well to begin with definitions of the usual quantities sought, as follows:

Refrigerating effect is the amount of heat abstracted by the ammonia from the cooling medium, as brine, expressed in B.t.u. per unit of time (minute, hour or 24-hr. day). This includes the waste cooling effect due to heat transferred from bodies it is not desired to cool—a waste necessarily ensuing from imperfect insulation, manipulation, etc.

The unit of refrigeration is a refrigerating effect of 288,000 B.t.u. per 24 hr., or 200 B.t.u. per min. To obtain this effect by ice at 32 deg. F., it would be necessary to melt 1 ton (2,000 lb.) of it in 24 hr., the latent heat of ice being 144 B.t.u. Units of refrigeration are therefore spoken of as tons more commonly than as B.t.u.

The capacity of a plant equals the number of units of refrigeration it delivers: That is, the number of tons of 32-deg. ice it would be necessary to melt per 24-hr. day to produce a refrigerating effect equivalent to that of the ammonia. Hence, this quantity is often referred to as "ice-melting capacity."

Ice-making capacity is sometimes considered in the case of plants devoted exclusively to the manufacture of ice. Expressed in tons per 24 hr., this is equal to between one-half and eight-tenths of the ice-melting capacity previously defined, the diminution being due to the losses in the process of heat abstraction by the brine, manipulations of cans, and to the fact that the water must first be cooled to 32 deg. and, after freezing, be chilled below that point. Rating in terms of ice-making capacity is therefore only definite when the temperatures of water and finished ice are stated.

The coefficient of performance is a more correct term for what is sometimes miscalled the efficiency of a refrigerating system. This quantity is the ratio of the refrigerating effect in B.t.u. per unit of time to the heat equivalent to the indicated work done by the compressor in the same time; that is, it is the useful effect divided by the power put in, and in this respect is superficially an efficiency. The term efficiency generally has reference to the relation of an energy, after passing through a conversion, to its value at the source. In the case of refrigerating machinery the compressor energy is *not the source* of the useful effect. In reality, it is the condensing water that does the cooling, the compressor being merely an auxiliary in the process. As the work done upon the ammonia is the paid-for item and as the refrigerating effect is the thing desired, it is useful to know the ratio of the two.

The ideal coefficient of performance is the maximum that could be obtained under a given set of operating conditions if there were no losses. It is fixed by the operating temperatures of condenser and refrigerator and is equal to

$$P_i = \frac{460 + T_r}{T_c - T_r}$$

in which the numerator is the absolute temperature of the refrigerator, T_r is its temperature in degrees Fahrenheit, and T_c is the temperature of the condenser in the same units. It should be noticed that neither refrigerator nor condenser is ever at one uniform temperature in operation. To maintain the heat flow, the brine must always be a little warmer than the ammonia it boils, and the circulating water always a little cooler than the vapor it condenses. But, assuming ideal apparatus, the ammonia vapor could be worked in the refrigerator up to the temperature of the outgoing brine (just as in a perfect steam boiler the steam might be worked up to the temperature of the outgoing flue gases), and in the condenser the vapor could be worked down to the temperature of the outgoing water. If these temperatures be used in the calculation, then, the resulting coefficient of performance represents ideal conditions of the whole system, including heat transmission of refrigerator and condenser. If, however, the temperatures corresponding to the vapor pressures of the ammonia are used, the resulting coefficient is that of the system, exclusive of losses in heat transmission in refrigerator and condenser. Numerical examples of these two values of the ideal coefficient of performance will be cited later.

The efficiency of the plant may be expressed as the ratio of the actual to the ideal coefficient of performance.

The economy of compression plants is often stated in pounds of refrigerating effect (ice-melting) per pound of coal, on the assumption that a certain number of pounds of coal are consumed to make one indicated horsepower at the compressor.

Other results should include the expense of condensing water (under certain conditions this may be as large as the cost of the fuel) and the cost of auxiliary power.

Only measurements of performance as indicated by the above-defined quantities come within the scope of this article. For complete tests, enabling a study of all the heat transfers, temperature and time-quantity determinations should be made of both ammonia and cooling water.

The duration of the trial should be at least 12 hr., and preferably 24, to allow for the possible error at the

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finish of the test due to a different amount of heat being stored in the refrigerator, condenser, etc., from that contained at the beginning.

Coming now to the measurement of the refrigerating effect, this, in B.t.u. per min., equals

$$R = W(T - t)C$$

in which

W = Weight of brine in lb. per min.;
T = Temperature of brine at inlet in deg. F.;
t = Temperature of brine at outlet in deg. F.;
C = Specific heat of brine.

For the determination of W the various methods of water measurement have been applied. Owing to the large quantity of brine circulated even in small plants, the method of direct weighing by tanks and scales is inconvenient. A modification of this consists in running the brine through two tanks, one above the other, the upper one having in its bottom a number of orifices through which the brine flows. The rate of flow varies with the height of the brine level in the upper tank. One of the orifices may be readily calibrated during the trial by collecting, weighing and timing its discharge at various heads. The total brine flowing through all the orifices may then be found by multiplication.

When meters calibrated in gallons or cubic feet are used, a separate determination of the weight of the brine per gallon or cubic foot is necessary. This may be made with a hydrometer, care being taken to test a sample withdrawn near the meter and at approximately the same temperature as it has in the pipe.

Concerning temperatures T and t, as the range is only between 5 and 10 deg., finely graduated thermometers must be used. For not more than 2 per cent. of error the graduations must be about 2 per cent. of the temperature range—that is, between one-fifth and one-tenth degree. The thermometers should be inserted in wells filled with mercury or oil and should be located in the inlet and outlet pipes as near as possible to the cooler. The protruding portions of the wells should be well insulated. During the trial it is well to interchange the thermometers as a check on their accuracy.

The specific heat of brine varies with its concentration, constituents and temperature. The last-named variation is comparatively small—about 0.06 per cent. decrease of the specific heat for each degree decrease in temperature. As to the effect of the constituents, the presence in calcium chloride brine of manganese and sodium chloride in moderate quantities (say up to 20 per cent.) does not affect the specific heat of the mixture materially. The effect of the concentration, however, is to lower the specific heat from unity (that of water) down to about 0.65 at a concentration corresponding to a specific gravity of 1.26.

The following formula for pure calcium chloride brine will give results for the specific heat of commercial calcium chloride brine to within 1 or 2 per cent. of error, between the limits of —4 and 40 deg. F., and specific gravities between 1.10 and 1.26:

$$C = 1.833 - 0.93G - 0.0005(32 - T_a)$$

In this formula[1] G is the specific gravity, and T_a is the average temperature of the brine in degrees Fahrenheit at inlet and outlet.

[1]Deduced from the values given in Bureau of Standards Bulletin No. 135.

When the cooling is done by direct expansion of the ammonia, it becomes difficult to find the refrigerating effect accurately. An approximation may be made, however, by finding the heat taken up by the condenser water and jackets and substracting from this the heat equivalent to the compressor work. The difference roughly equals the refrigerating effect.

Ice-melting capacity is readily calculable from the value of the refrigerating effect R in B.t.u. per min., by dividing by 200, the result being in tons per 24 hr.

To obtain the actual coefficient of performance, it is necessary to indicate the compressor. The process here is similar to that for steam-engine trials and hardly needs comment other than that a special steel-lined indicator must be used with as short pipe connections as possible to avoid materially increasing the clearance space. Having obtained the indicated horsepower of the compressor, the coefficient sought is

$$P_a = \frac{R}{42.4 \times I.hp.}$$

The ideal coefficient of performance may be found from the average thermometer readings at the brine and condenser water outlets, as previously defined. Or, if vapor pressures are used, the corresponding temperatures may be found from the ammonia tables. When the plant is equipped with pressure gages bearing thermometric scales, the ammonia tables may be dispensed with.

As an illustration of the difference that may exist between the values of the ideal coefficient as obtained by the two methods previously discussed, consider the following data:

Suction pressure, 28 lb. gage; from tables, temperature, 15 deg.; head pressure, 151 lb. gage; from tables, temperature, 84.5 deg.; outgoing brine temperature, 28.9 deg.; outgoing condenser-water temperature, 83.5 deg.

From these figures the ideal coefficient of the whole system is

$$\frac{460 + 28.9}{83.5 - 28.9} = 8.95$$

and for the system, exclusive of condenser and refrigerator transmission losses, is

$$\frac{460 + 15}{84.5 - 15} = 6.83$$

The measurement of the condenser water is simpler than that of the brine, the quantity being much less. Direct weighing or the usual forms of water meter may be employed, care being taken to select one appropriate to the conditions of flow, whether pulsating or steady. The number of gallons of cooling water per minute per ton of ice-melting capacity should be figured for purposes of comparison and also in order to add the cost of this item to the other costs to obtain the total.

The performance of auxiliaries may be either estimated or measured. When they and the compressor are steam driven, it is well to measure the total steam supplied to the plant by the boiler-feed method or by a steam meter installed so as to avoid inaccuracies due to pulsating flow. The steam consumption of the compressor and of the whole plant may then be stated in pounds of steam per ton of ice-melting.

Ratings are generally based on head and suction pressures of 185 and 15.7 gage respectively. Any departure from these conditions should be considered in making comparisons of test results.

Large-Capacity Tank-Type Circuit-Breakers

The main changes that have been made recently in tank-type oil-break circuit-breakers have been introduced to make the breakers more accessible for inspection and repair. No radical departure in the design of the current-carrying or operating parts has been necessary.

Indoor and outdoor breakers manufactured by the General Electric Co., Schenectady, N. Y., are practically similar. The only difference consists in the addition to the indoor breaker of a few parts to enable it to be serviceable both from a mechanical and an electrical standpoint under all weather conditions.

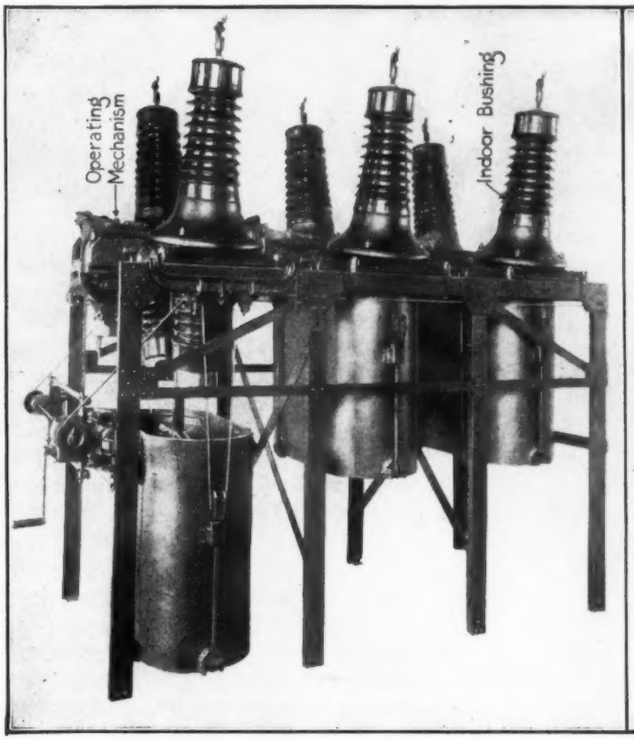


FIG. 1. INDOOR HIGH-VOLTAGE OIL SWITCH

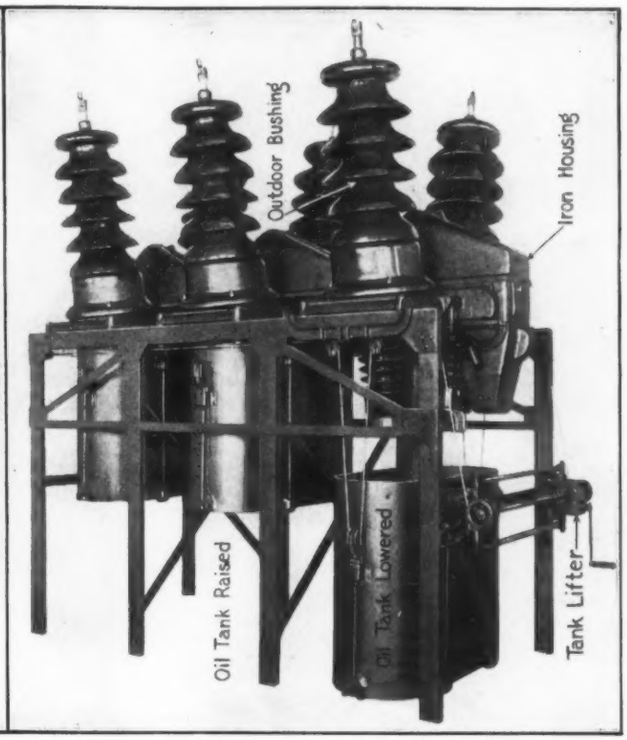


FIG. 2. OUTDOOR HIGH-VOLTAGE OIL SWITCH

The oil-break circuit-breaker, shown in Fig. 1, is used indoors on voltages from 35,000 to 150,000, and that in Fig. 2 is used on outdoor work from 22,000 up to 150,000 volts. The breakers shown are designed for 70,000 volts and 300 amp. They are alike except that the former has outdoor bushings and the mechanism is protected from the weather by an iron housing for outdoor installations.

A noteworthy advance in the breakers is mounting them on framework and handling the tanks by a tank-lifting device recently perfected. The lifter consists of a detachable frame with shaft, handle, worm gear, winding and unwinding drums. The advantage of this equipment is that it allows a tank to be removed or placed in position without difficulty, as shown in the illustrations. The device is readily detached and can be moved by one man from one switch to another, thus making it a simple operation to lower or raise an oil tank and also rendering it easy to align, inspect contacts and oil and replace contacts if necessary.

These top-connected circuit-breakers are self-contained for use on systems of potentials up to and including 200,000 volts. One tank with two breaks in series is used in each phase. The circuit-breakers are made for either automatic or nonautomatic operation. They are closed by hand, solenoid or air, and those of the automatic type are tripped under overload by series trip coils (mechanical trip indoor only) or current transformers. In using these circuit-breakers for hand operation they can be installed behind the switchboard panel with the operating handle in front and connected to the breakers by connection rods and bell cranks. All can be mounted on a framework, except those for 110,000 volts and above.

The operating mechanism and bushings are secured to the cast-iron cover of the heavy sheet-steel tank. Each bushing extends through the switch cover into the oil and is so clamped to the cover that the throwing of oil is impossible. The bushings are easily removable and interchangeable.

There are two fixed contacts in each switch element between which one phase of the circuit is made and broken by a horizontal contact blade. Each contact blade is connected to the operating mechanism by a specially tested hardwood rod, which passes through the cover of the switch in an insulating bushing. The stationary contacts consist of widely flared fingers and long arcing tips of drop-forged copper secured to the contact block by flat-steel springs, copper laminations and screws. The contacts are always smooth and bright, due to the sliding effect they are subjected to on opening and closing and the arrangement of burning tips.

The oil furnished for these breakers has a very high flash point and high resistance to carbonization. The insulating value of the oil and its burning and carbonizing characteristics make a difference in the length of time the arc will hold.

Ideal Ventilation never permits sharp drafts across a room. The air should be abundant, but so diffused as not to be distinctly felt at any occupied point.

Traveling Screens at Stations of Philadelphia Electric Co.

By Henry J. Edsall*

SYNOPSIS—Description of water screens at Christian Street stations, which are designed to take care of over four hundred million gallons per twenty-four hours.

In pumping water from a river or other source of supply through the small tubes of a surface condenser, it is necessary to use some sort of a screen that will effectively keep out trash, fish, etc., that might clog the tubes. Even a leaf or small piece of paper over the end of a tube may stop the flow of water through the tube.

In the past it has been the practice to use stationary screens, which could be raised and lowered—usually two screens, one ahead of the other, so that if one of them became clogged with trash, it could be raised and cleaned and the other would do the screening meanwhile. Under ordinary conditions of the water the labor necessary for cleaning the screens is not a serious matter, but in the spring and fall and after heavy rains the water is apt to contain a great deal of trash, so that the cleaning of the screens becomes laborious and if they are allowed to become clogged with trash, the flow of water is decidedly reduced as well as the efficiency of the condensers.

In order to eliminate these difficulties, it is fast becoming standard practice to use, in place of the fixed screens, some type of traveling screens arranged so that at some point in their travel they are automatically cleaned of trash. The type of screen that has been adopted consists of two strands of long-pitch chain, to which are attached steel frames carrying the screen wire, the width of these frames being the same as the length of pitch of the chains, so that there is a screen for each chain pitch. The frames coming close together form a continuous screen apron. The chains run over sprocket wheels at the head and foot, the foot being under water and coming close to the bottom of the forebay and the head being somewhat above the operating floor. Each screen has its own compartment in the forebay, and they are made of the proper width to come close to the side walls, so that the water cannot get past on the sides or at the bottom. In connection with the screen frames are small ledges, or buckets for carrying up the trash, part of which is discharged as the screens turn at the head wheel and the balance cleaned off by a water spray with considerable pressure behind it, which is arranged to come from the inside of the screen and wash the trash off the outside just after the screens turn and start to descend. The travel of the screens is at an angle of about 60 deg. with the horizontal, so that the trash that is dumped off at the head can be dropped into a trough underneath the delivery point. The amount of condensing water required to pass through the screens at stations A-1 and A-2 (Christian Street) of the Philadelphia Electric Co. is 411,000,840 gal. in 24 hr. when running under full load in summer. In order to provide for this amount of water, six traveling screens were decided upon, each screen apron having an effective width of about 5ft., making a total width of screening surface of about 30 ft. The center of the footshaft of each screen apron is 31 ft. 11 in. below the operating floor, the bottom of the intake discharge openings being just about at the same level, and the size of the intake and discharge openings 6x5 ft. The chains are 18-in. pitch steel strap. The pins at the joints are of ¾-in. cold-rolled steel with the ends milled off flat, fitted into the links and riveted over, thereby holding the pin rigidly in the links. In between the inside links at each joint there are cast-iron rollers with tobin-bronze bushings, which run on the steel pins. When the screens are in operation, the rollers travel up and down on guides, thereby supporting the screen apron. The sprockets at the head and foot are 7-tooth wheels 41½ in. diameter, all keyed to the shafts, which run in babbitted bearings, no provision being made for oiling the foot bearings under water, as this does not seem to be necessary. The shafts are of nickel steel in order to avoid corrosion as far as possible. Each screen headshaft is driven by means of a chain drive from a countershaft, the latter connected through a reducing-gear arrangement to a 5-hp. motor.

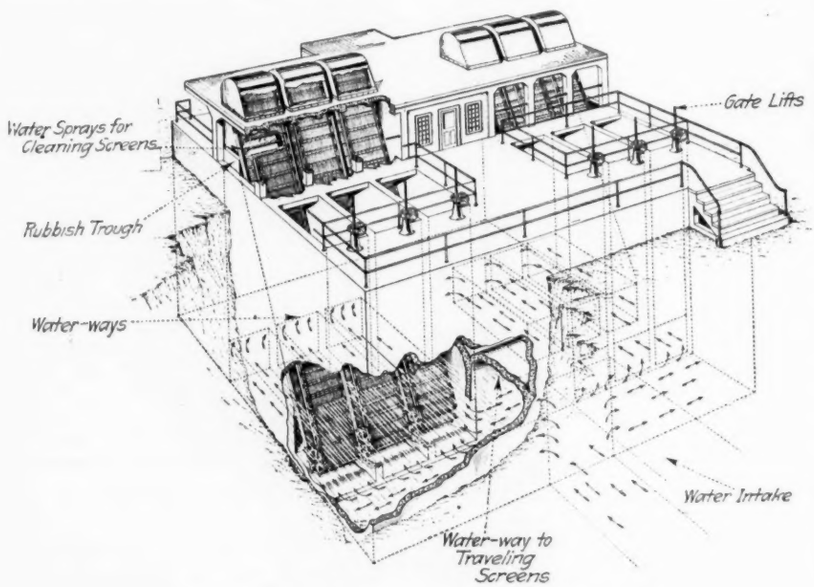


FIG. 1. GENERAL ARRANGEMENT OF SCREENS

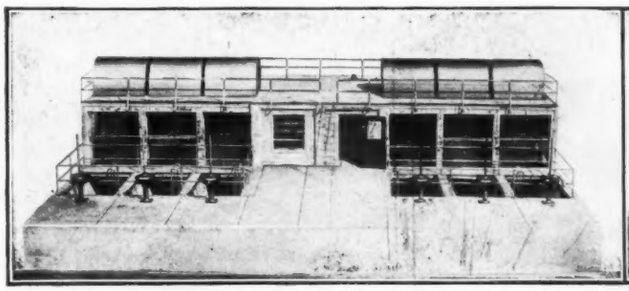
The screen frames are made of ⅜x2-in. steel flats bent into rectangular form and welded together, the screen wire being flanged up at the edges and bolted to the frames by means of ½-in. bronze bolts with steel

*Engineer with the Link-Belt Co.

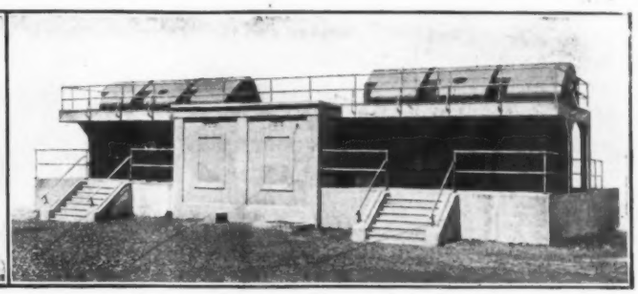
washers under the nuts. The screen wire itself is No. 10 B.w.g. galvanized wire ½-in. mesh.

The forebay is built almost entirely of concrete, the bearing supports at the head being attached directly to the concrete and the heads of the screens being covered by steel hoods. The six screens are placed in sets of three each, with a motor house in between. Each screen compartment has a gate valve on both the intake and outlet sides of the screen, so that the water can be shut off from any screen in case it is desired to pump out one of the compartments to get at the lower end of the screen. The spray pump for delivering the spray for cleaning the screens is located in the concrete motor house and has a capacity of 500 gal. per min. It operates the spraying nozzles on three screens at a time.

The screens are operated by one man, and when the water is comparatively clean, it is necessary to clean them only about every eight hours. As the water gets bad, they are operated oftener. Their travel is about 50 ft. per min.



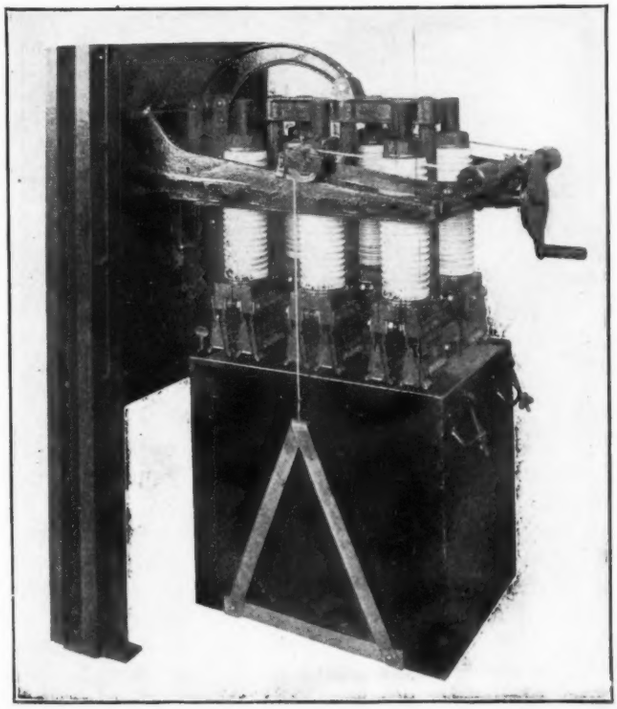
FRONT VIEW OF SCREEN HOUSE



REAR VIEW OF SCREEN HOUSE

Tank Lifter for Oil Switch

The General Electric Co., of Schenectady, N. Y., has recently developed a number of tank-lifting arrangements for oil switches of various forms and sizes. The lifter herewith illustrated is employed for switches of capacities up to 1,500 amp. The simplicity of construction is evident from the illustration. The oil tank can be lifted the entire distance between the switch frame and the floor.



OIL TANK PARTLY LOWERED BY TANK LIFTER

The method of removing an oil tank is as follows: The lifter is placed on the oil-switch frame and fastened there by turning two wing-nuts; the operating handle connected to a worm gear is turned until the two triangular tank supports engage the tank, which is then unfastened from the switch frame and lowered to the floor or to any intermediate position desired.

The lifter is adapted to various sizes of the form of switch shown simply by moving slightly forward or back the pulleys over which the cables run, depending upon whether the switch on which the lifter was last used is smaller or larger than the switch next to use it. The device is readily detached and may be moved from one switch to another by one man.

Fusible Plugs for Marine Boilers—Amendments adopted by the Board of the United States Steamboat Inspectors on Mar. 31, to its rules relating to fusible plugs, were as follows: Fusible plugs shall be "filled from end to end with tin not less than 99.5 per cent. pure," in lieu of 99.7. Fusible plugs shall be renewed at each annual inspection, but where they were installed or renewed not more than six months prior to the annual inspection, they may be allowed until the next following annual inspection or for a period not to exceed 18 months. In such cases, however, the inspector shall satisfy himself at the annual inspection that the plugs are in condition to warrant their continued use until the next annual inspection, after which they shall be renewed at each annual inspection. Manufacturers of fusible plugs shall furnish the Supervising Inspector-General a sample plug from each heat for examination and test, but where more than 250 are poured from the same heat a sample shall be furnished for each 250 plugs or fraction thereof. The samples furnished shall bear the same number from any one heat.—"International Marine Engineering."

Millions of Cords of Wood Wasted Annually—There are more than 48,000 sawmills in the United States, and their by-product in the form of sawdust, shavings, slabs and other wood refuse is estimated as 36 million cords per year, one-half of which, perhaps, serves a useful purpose as fuel under the boilers. Much of the remaining 18 million cords not only is not utilized, but generally is a source of danger and costs the mill time and money to dispose of it in various ways. Some goes to the local fuel market, some to pulp mills or to wood-distillation plants, while shavings and hog cuttings and other mill waste is sometimes used to fill low places in the yard, but the most common way of getting rid of waste is by burning. It is estimated that for a mill of one hundred thousand feet capacity the cost of conveying the waste and destroying it in a closed burner is 42c. per cord, or $10.05 per day, and it seldom brings in any revenue. The production of waste is unavoidable, so the next best thing is to seek out some way of utilizing it so that it will pay for its disposition. The Forest Service is working on this problem, but has not yet found a satisfactory solution.

York Poppet-Valve Return-Flow and Uniflow Engines

Under Rosenzweig patents the York Manufacturing Co. is now building poppet-valve engines of the return-flow and uniflow types. In both designs simplicity and accessibility of cylinder and valve-gear parts has been the watchword. Simplicity has been attained as the accompanying illustrations will testify, and at the same time the economy has not been overlooked. Clearance spaces and volumes have been held to a minimum, and the use of a combined relief and expanding poppet valve prevents leakage of steam. The valves are horizontal and are operated by cams. A single eccentric mounted on the layshaft carrying the governor operates the steam valves, operated by cams acting on antifriction rollers, effecting a rapid opening and closing. The cams are clamped on the shaft and can be shifted into any position to give the desired release and compression.

The steam valves are operated by oscillating levers provided with rollers and curved cam pieces attached to the valve spindles. The oscillating levers for both valves are mounted on a single shaft, which receives its motion from an eccentric mounted on the layshaft. It is noteworthy that this arrangement of the valve gear necessitates the use of but one eccentric.

Supported by long spindles and guides, the valves are

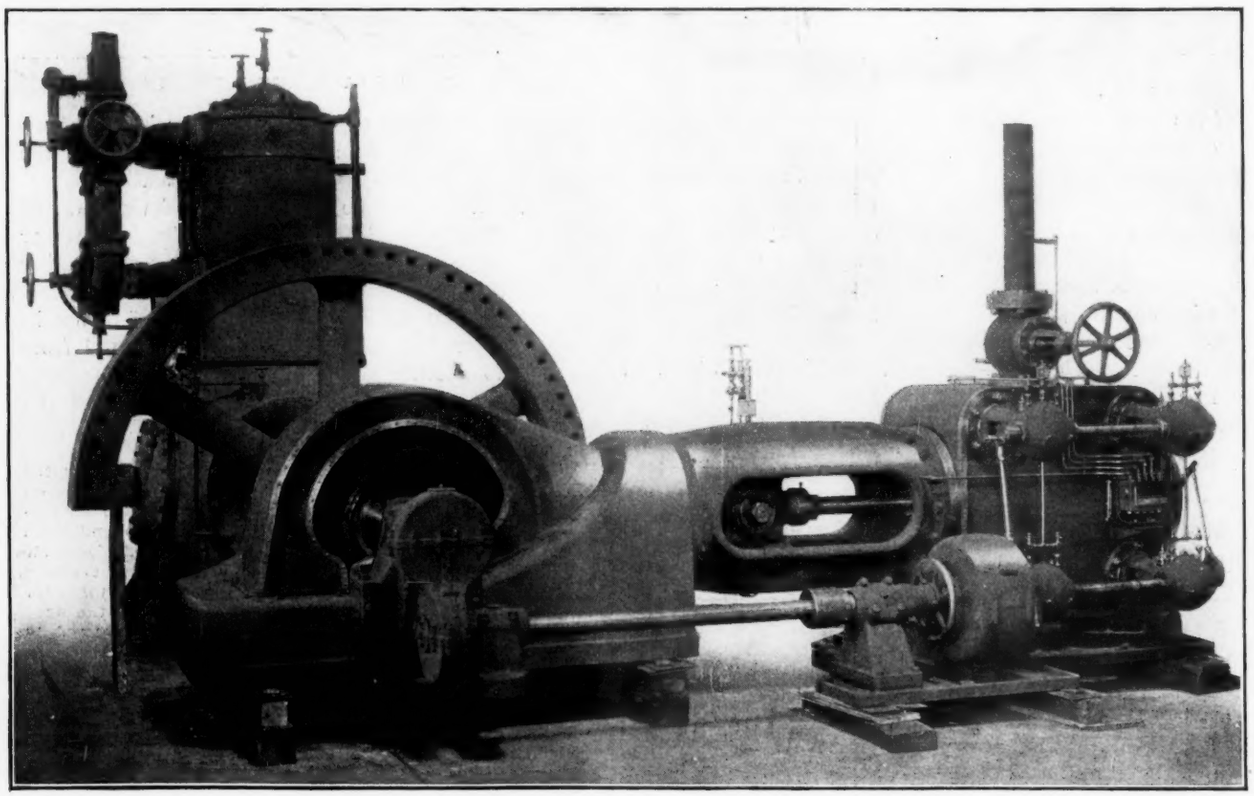


FIG. 1. YORK POPPET-VALVE RETURN-FLOW ENGINE DRIVING VERTICAL COMPRESSOR

while the cams controlling the exhaust valves are clamped directly on the layshaft. This arrangement greatly simplifies the valve gear. As the valve gear is inclosed and runs continuously in oil and the layshaft has a drag-crank connection, the operation of the engine is practically noiseless. A more detailed description of these features follows:

The valve gear is operated by a layshaft running alongside the engine and driven from the mainshaft through a drag crank and shaft, by means of a pair of spiral gears. The drag crank is self-adjustable and prevents unavoidable motion and jars of the mainshaft being transferred to the gears, which for this reason run noiselessly and are not subjected to undue wear.

The layshaft is placed in line with the exhaust valves, the bonnets of the exhaust-valve gear being provided with bearings to support the shaft. The exhaust valves are placed in a horizontal position. The pressure on the guide, neglecting the spindle, amounts to about ½ lb. per sq.in. The spindles are lubricated by means of force lubrication which passes the oil along the valve spindle through spiral grooves, on to the guide and then into the steam space, thus obtaining a triple effect besides preventing an escape of steam. The valve spindles are ground into long bushings and provided with a labyrinth packing consisting of small circular grooves.

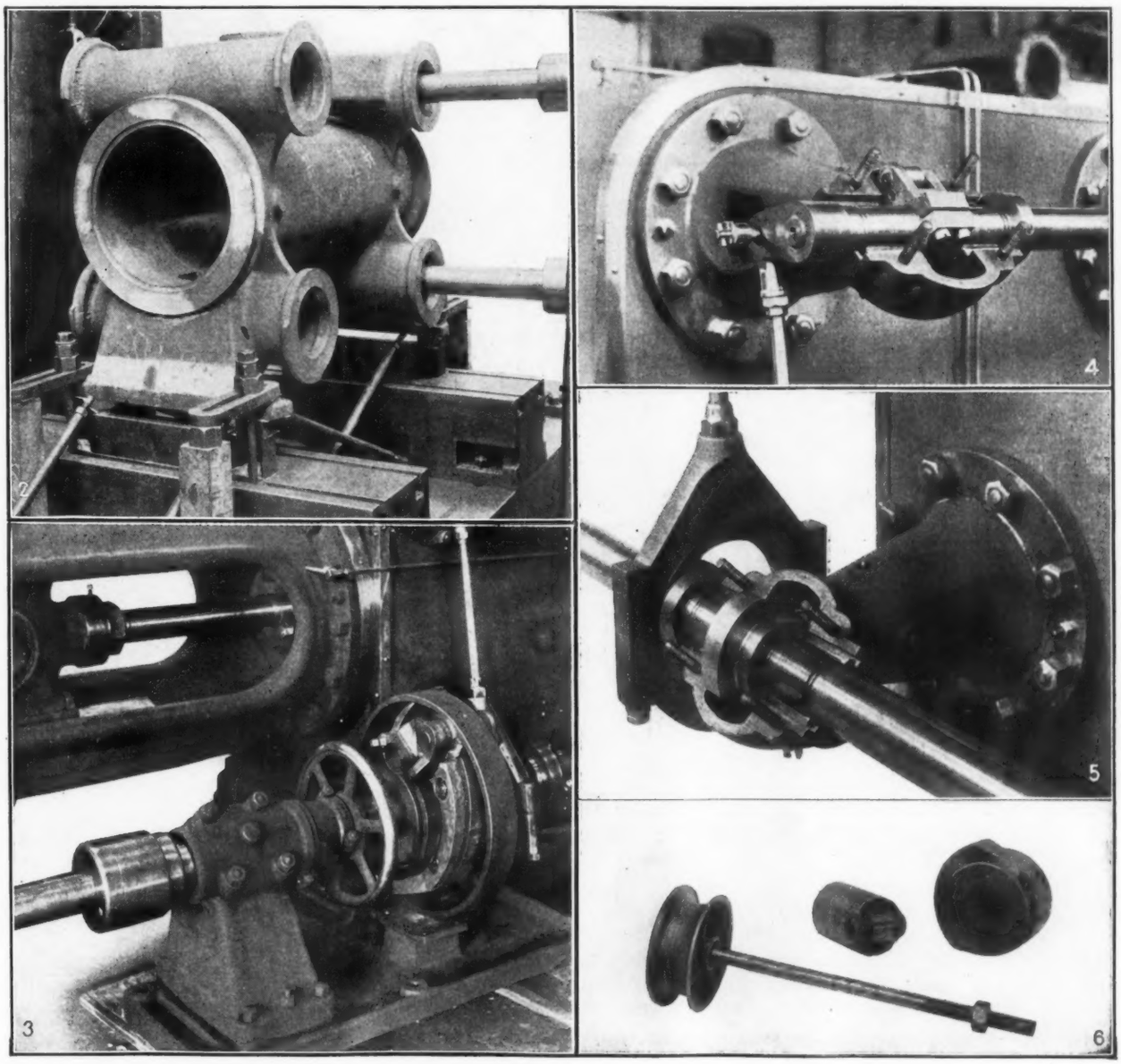
Of particular interest is the design of the valve cages, which contain not only valve seats and spindle guide, but also the complete structure of the ports. The whole valve assembly is accessible by removing the back covers without interfering with the valve gear. The valve-gear parts work in a removable cover, preventing the ingress of dirt, and operate in an oil bath through which the oil may be circulated continuously and used over and over again.

Different load conditions of operation are met by the use of a compound eccentric operating the steam valves. A small eccentric is keyed to the shaft and a larger one arranged slidably on it and connected to and controlled by the shaft governor. The large eccentric is so shifted that the combined eccentricities of the two eccentrics become such that practically a constant lead is maintained, while the throw increases according to the requirements of the load.

The shaft governor on the engine under discussion provides large centrifugal forces which make it sensitive and powerful to withstand the backlash of the valve gear. It is completely balanced and consists of two pendulums which by means of short arms shift a sleeve mounted on the shaft. This sleeve in turn is connected to and shifts the large eccentric. While the throw of the pendulum is large, the travel of the circular spring of moderate cross-section is short, producing a governor of good regulating qualities. A hand speed adjustment is provided which allows a change in the speed while the engine is in operation, an increase of 50 per cent. being possible.

To lubricate the eccentrics, eccentric strap and governor, the oil is introduced into the hollow layshaft and forced by centrifugal action to the desired points. A force-feed lubricator introduces the oil into the valve spindles and cylinder, and sight-feed oilers are used for the valve gear.

The arrangement of the uniflow engine is essentially the same as the return-flow engine. The steam valves are operated and regulated in the same way. A central exhaust port controlled by the piston is provided for the escape of the steam. As this port is of unusually large size, about three times as large as the valves and ports provided in return-flow engines, the release and exhaust of steam takes place rapidly, and the pressure in the cylinder will be as low as in the exhaust pipe or condenser.



FIGS. 2 TO 6. VIEWS OF PARTS OF THE YORK ENGINE

Fig. 2—One-piece cylinder on boring mill. Fig. 3—Centrifugal governor and handwheel for speed adjustment. Fig. 4—Oscillating lever and roller operating steam valve. Fig. 5—Exhaust valve cam on layshaft. Fig. 6—Exhaust valve and operating mechanism

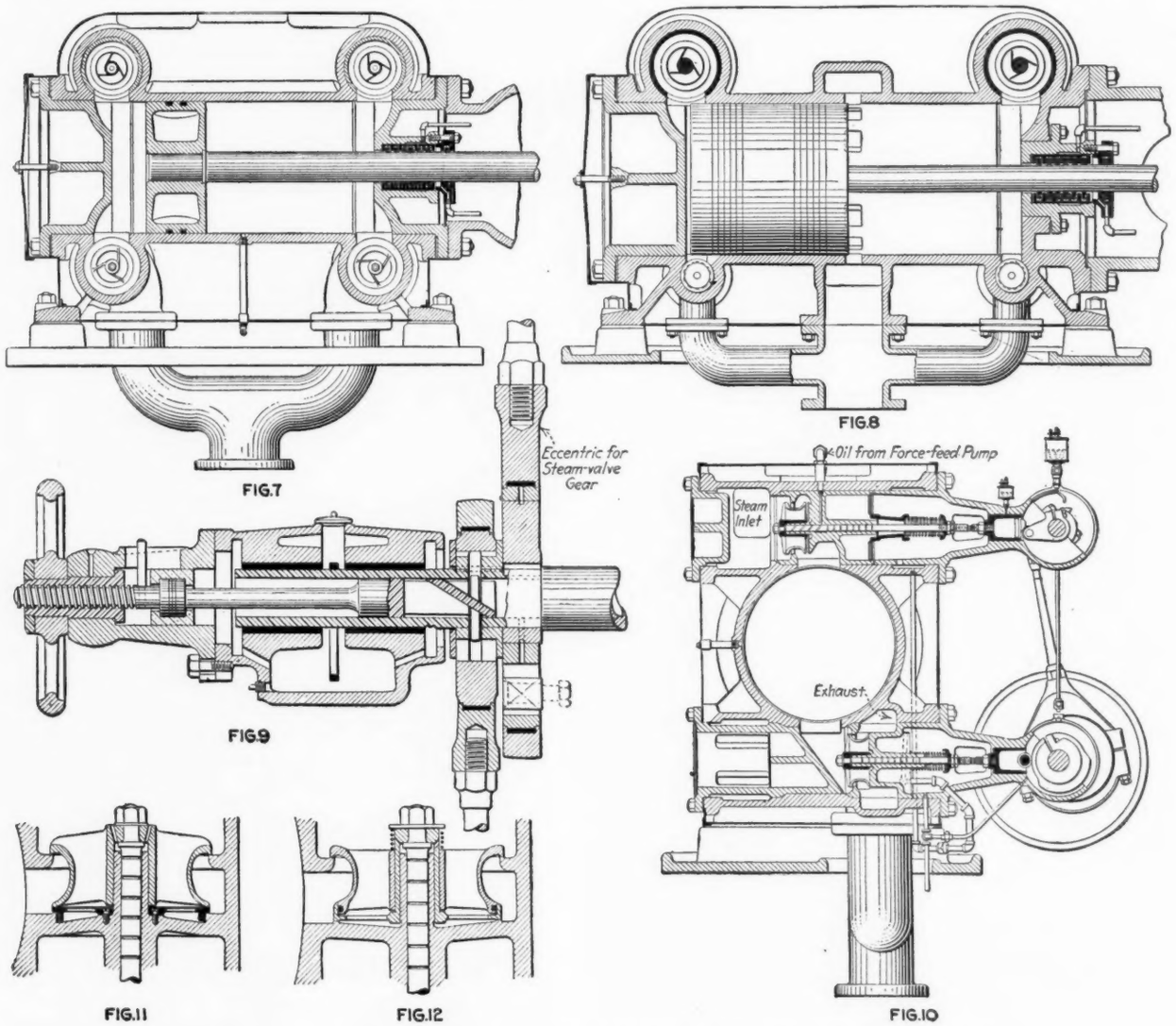
In the standard type of uniflow engine, compression takes place during approximately 90 per cent. of the stroke. A high compression is advisable and economical in this type of engine, but the degree should depend upon the point of cutoff, the vacuum and other working conditions. With this point in view, auxiliary exhaust valves and gear controlling the compression for all operating conditions were provided.

After the main exhaust port is closed by the piston, steam is allowed to pass through the auxiliary exhaust valves into the exhaust pipe. This steam travels at low velocity and is free of moisture, being relieved of pressure and water when the main exhaust port opens, hence has little cooling effect on the cylinder cover and walls.

The auxiliary exhaust valves are of small diameter, single-seated, yet balanced as they open under no difference in pressure on either side of the valve.

These valves are operated in a way similar to the steam-admission valves; but in this case the eccentric is arranged slidably on a block which is keyed to the shaft. It is obvious that the slide block can be so set in relation to the eccentric and steam-valve gear that the auxiliary exhaust valves begin to open always at the same predetermined point, say approximately at dead center; the eccentric can then be moved either automatically or by hand into any position on the block to effect closing of the valve at any point of the stroke.

Instead of having the eccentric controlled by a governor, a shifting device operated by hand while the engine is in operation is provided, which makes it possible to move the eccentric to any position and thus obtain any desired degree of compression. This feature is also valuable when starting, as the compression may be temporarily reduced.



FIGS. 7 TO 12. SOME DETAILS OF YORK RETURN-FLOW AND UNIFLOW ENGINES

Fig. 7—Return-flow poppet-valve engine cylinder. Fig. 8—Section through uniflow engine cylinder. Fig. 9—Variable eccentric for controlling compression. Fig. 10—Transverse section of return-flow engine cylinder. Fig. 11—Poppet valve with flexible seat. Fig. 12—Combined relief and expanding poppet valve

One of the difficulties in poppet-valve engines is to design and manufacture the valve and seat so that they expand equally and prevent leakage of steam under any other temperature than that at which they were ground together. The York poppet valve is made so that it will adjust itself to any expansion of the valve seats. The upper half will lift under an excess of pressure within the cylinder or in case of a slug of water, and thus constitutes a relief valve. This part lifts only against steam pressure and a spring of light tension and is in no way affected by the tension of the main spring.

Editorials

Power Plant Tendencies

An interesting example of the tendencies of the times toward large units of all kinds—boilers having two or three times the average heating surface of former years and turbines of large capacity—is presented in the leading pages of this issue. Although the boilers in size do not approach those at Delray or at the new plant of the Ford company, they are large even when compared with present-day standards and possess the usual advantages of the large unit—lower initial cost, less labor to operate and a reduction in maintenance if the furnaces are of proper design. The latter in the present case have been made unusually high to provide large combustion volume so essential when the boilers are to be forced much above their normal rating.

A four-pass Stirling boiler is not common, but in the present case the baffling works out to good advantage, as the economizer has been placed on the floor adjacent to the gas outlet from the fourth pass. With a large free opening and no interconnecting flues, the resistance to the flow of gases should be reduced to a minimum. In this location the economizer is within easy reach of the operator and in a more comfortable place for inspection or repair than a position above or near the tops of the boilers. Flanged connections between the water and tube headers, permitting the blanking off of the latter without interfering with the operation of the boiler, is an innovation worth while, that should be duplicated in future installations.

With the condenser auxiliaries motor-driven and exhaust steam available only from the turbine-driven feed pumps and the stoker engines, the economizer has a wider range and greater possibilities of effecting a saving. Following standard practice the tubes and headers are of cast iron. They have withstood pressures of six hundred pounds before and four hundred and fifty pounds after installation at the plant, although the working pressure on the boilers is two hundred pounds. Some doubt has been expressed as to the desirability of using cast iron for the higher pressures, but in other plants the same metal has been used in economizers for pressures in excess of that mentioned.

One feature of the plant that should attract attention is the simplicity of the coal-handling equipment. A traveling crane and grab bucket transfers the coal directly from the railway car to large stoker hoppers. There are no intervening conveyors or bunkers, although a pit beside the track and in front of the boilers provides ample storage capacity. With no overhead work or complication of conveyors, the initial cost should be low, and by reason of the simplicity of the equipment the same should be true of the maintenance and operating charges.

The plant in its entirety is interesting as an example of modern construction pointing toward the tendencies of the times: Large units, high pressures, electric-driven auxiliaries and extended use of the economizer.

Factory Visits by Engineers

Anything that can shorten the line of communication between the user of modern power-plant machinery and the manufacturer deserves serious consideration. This is wholly apart from the desirability of frequent visits by consulting engineers to the factory on behalf of clients. It is not suggested that the consulting engineer shall be debarred from performing his useful services nor that expense accounts for mere junkets shall be "O.K.'d" by employers of operating men. There must be an exchange of useful ideas if the factory visit by the operating engineer is to pay. The immediate opportunity is not so much along the line of better plant development as in bringing to the manufacturer more specific information as to the detailed routine and emergency performance of his product in service; not under test conditions, but under the severe requirements of everyday commercial operation.

When one gets down under the skin of the average intelligent user of machinery, it is possible to learn a great deal about one's product that does not appear upon the surface. When new and radical equipment designs are placed in service, manufacturers naturally follow performance quite closely; but after apparatus has become fairly standardized, other problems are apt to come along, and in larger development work the fitness of various details for year-around service may be overlooked. There is no question that in many plants where less than the expected results is gained from subordinate equipment, let alone the major units, a visit to the manufacturer, if feasible, might prove most helpful. No one else can realize the actual good and weak points of a piece of machinery so well as the man whose daily duty it is to supervise its operation, assuming of course that a high-class operating man is in command. Yet how few engineers ever have the chance to visit the shops where the machinery is produced for their use. Many operating men are acquainted with the local or district representative of the manufacturer, but such an acquaintance, even if it develops into a real friendship, is not wholly a substitute for a downright "shop talk" with the designing and manufacturing engineer on his "home grounds."

A visit of the kind outlined here may be made a real opportunity for both parties. The operating man may be expected to face a lot of keen inquiry about his practices and must naturally be prepared to give as good as he seeks. On his part, he should be primed to defend his ideas and not be in the least afraid to make suggestions leading to the modification of designs or of manufacturing methods, if it appears to him that these will result in better service. It will help such a man to learn something about the necessities of multiple production, of standardization in the factory process, and of economy in labor and material from the shop standpoint. The trouble is that too often in the past the operating engineer has not appreciated the difficulties the manufacturer is facing,

and neither has the manufacturer, in many instances, realized the minor trials which the user of his product is passing through. Like many another difficult situation that can be solved by bringing together the right men, the more efficient application of machinery in power-plant service can be realized by a greater degree of intercourse between maker and user; and it is the best estimate in the world that if men of these diversified viewpoints meet now and then upon the common ground of experience and frank discussion, a new plane will be reached in many branches of power-plant practice.

Distribution Losses

In the generation, transmission and distribution of electrical energy it is important that the losses be kept down to a value that is compatible with good design. Since electricity is the most convenient form in which energy can be transmitted, it may at first thought seem that the transmission and distribution losses would be small in well-designed systems. A careful analysis of this subject shows the contrary to be true. On the average alternating-current system the distribution losses amount to from fifteen to twenty-five per cent. of the total power delivered to the busbars. About fifty per cent. of these losses are transformer-core losses, thirty-five per cent. line transformer-copper loss and leakage, and fifteen per cent. meter-shunt losses.

Significant in the division of these losses is the large proportion creditable to the transformer cores and meter shunts, showing the importance of keeping the all-day losses, even if they are small, down to a minimum. On systems where an all-day service is given with many small transformers and meters, the energy lost in distribution may amount to fifty per cent. of the total transmitted.

The line losses form a small percentage of the total, and this would seem to indicate that they do not have a very important bearing on the operation of the system. On lines that are supplying mostly a lighting load, a large percentage of these losses takes place in a period of about three or four hours; consequently they may be comparatively high during this period, yet when compared with the other losses that are going on for twenty-four hours per day, may seem to be low. The line losses directly affect the voltage at the consumer's meter, and if they are high, the voltage drop at the service will be high, which means a loss of revenue to the central station and an unsatisfactory service to the customer. Among the chief causes of complaint in the lighting service are flickering and dim-burning lamps; therefore it is essential that the line losses be kept small or that the voltage be maintained constant at the receiver by automatic voltage regulators.

In this issue is an article on "Distribution-Loss Factors," in which the author discusses the losses involved in the transmission and distribution of electrical energy and by reference to actual working conditions makes some very interesting comparisons. These show a wide variation in the losses for different systems, and as might be expected, the higher losses prevail in the small plants. However, in many cases this is not due to inefficient operation or poor design of the small plant, but to a load factor inferior to that of the larger installations.

Diagnosing Trouble

Prompt resumption of service and a clear understanding of the fundamental causes of operating difficulties demand more systematic study than is customary on the part of many engineers. There is nothing new in the idea that more is to be learned from the troubles of apparatus than from its successful routine service, but the lessons of the breakdown, complete or partial, are worth going into at pretty much any cost. Through fear of giving offense and the disinclination of many engineers to enter into controversies, the frank discussion of equipment difficulties is often hard to maintain in conventions, but such discussions could well be given right of way at local meetings of engineering organizations.

A good hobby for the coming winter season for engineers with the analytical type of mind would be the accumulation of all data as to accidents, failures and mishaps small and large, of equipment either under their charge or within reach as to facts.

Too many times a man fails to put down the reasons for erratic behavior of his apparatus, merely noting on the log sheet that Unit No. 2 was out for fifty-five minutes on a given date, without attempting to specify the cause. In most cases it would be absurd to expect an operating man to start a course of reading on a given machine trouble before taking steps to maintain service otherwise, and the notation of causes of mishaps or faults is not expected to take the place of quick thinking and accurate reasoning in emergencies. Nevertheless, the diagnosis of operating troubles may be greatly assisted by more care in putting down the lessons of daily routine and emergency service that arise from irregularities in performance.

By the preparation for practical service of a sort of personal "manual-at-arms" bearing upon the troubles, symptoms and remedies of power-plant equipment, the engineer may enter a field of analysis both interesting and profitable. Equipment is becoming too costly to depend wholly upon the rule-of-thumb class of "trouble-shooting" practitioners. Faithful study of the causes of operating difficulties in times when things are moving smoothly will go a long way toward preparing one to deal intelligently with emergencies when they arise.

"When scientific work is solely with the object of securing commercial gain, its correlative is selfishness; when it is confined to the path of narrow specialization it leads to arrogance; and when its purpose is materialistic domination, without regard for the spiritual needs of humanity, it is a social danger and may become an excuse for learned barbarity."—Professor R. A. Gregory in his new book, "Discovery, or the Spirit and Service of Science." An excellent thought for engineers of all classes to ponder over.

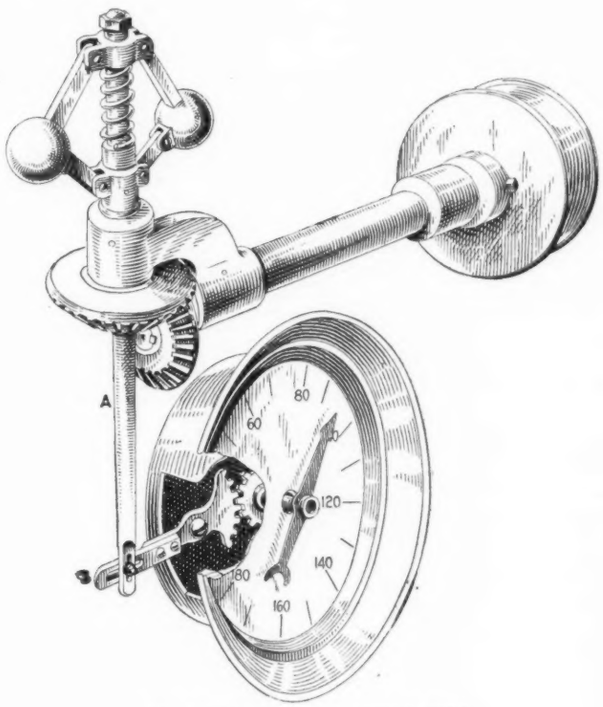
If it was "seeing the wheels go 'round" that formerly made the engine room so much more interesting than the boiler room, that day is passing, for there is not much fascination about a steam turbine so far as visible motion is concerned. Perhaps this will act to bring the boiler plant to the point of receiving the attention that it deserves.

Correspondence

New Duties for Old Equipment

An improvised speed-indicating gage was made as follows for attachment to variable-speed shafting, where it was desirable to observe and to alter the speed in accordance with differing requirements:

A small governor, taken from a discarded high-speed engine, is belted to the shafting. The valve stem *A* is



SHAFTING SPEED INDICATOR

lengthened to reach the lever *B* of an old steam gage, as shown, and a portion of the gage casing is cut away to accommodate the changed mechanism. The pressure tube is eliminated. The lever *B* is made adjustable as to length, so that by changing it the gage is set in proper ratio, making the original dial graduation correspond to the speed of the shafting. H. K. Scholefield.

Kohala, Hawaii, H. T.

Wedge Adjusting Bolts Broke

Several years ago, upon taking charge of a textile-mill power plant, the assistant engineer stated that the wedge-block adjusting screw on the crank end of a 22x42-in. engine broke about every two months during the three years it had been installed, and that a break was about due.

It happened about two weeks later, when there was a sharp noise as though something had been thrown onto the oil casing. Investigation showed that the head and about an inch of the lower wedge-block screw had dropped into the engine base.

When starting to replace the broken screw it was found necessary to take out the back half of the crankpin box in order to remove the wedge block and then there was difficulty in getting out the 2 in. of broken screw.

I took two new screws and sawed a slot in the end of each so that when one broke in the future a screw-driver could be used to take it out; and that is what I would suggest be done with all such bolts, so that when one breaks it can be easily removed with a screw-driver in a few minutes and several hours will not be wasted dismantling the oiling apparatus, etc. I would also suggest as a progressive idea that manufacturers who use this type of adjustment on engines or pumps do the same.

These screws usually break flush with the end of the wedge block and is probably caused by a slight movement at the end of the stroke, causing the screws to crystallize and break. I later had $\frac{1}{32}$ in. turned off these screws from the head as far as they extended through the solid part of the connecting-rod and have not had a broken bolt since. John M. Coleman.

Somerville, Mass.

Action of Venturi Meter with Pulsating Flow

While using a venturi meter to determine the amount of air used, the fact that the water column in the manometer that indicated the "venturi head" stood at about eight inches when no air was being used caused considerable speculation as to the reason and skepticism as to the accuracy of the device for measuring air flow.

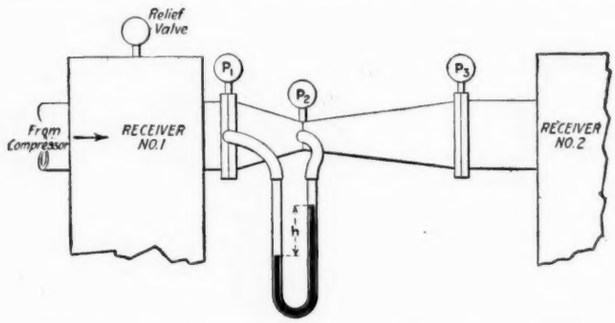
The set-up consisted of a two-stage, cross-compound air compressor, delivering air to a large receiver fitted with an adjustable relief valve. The air used in the tests passed from the receiver through the venturi meter into another smaller receiver and thence through temperature-regulating apparatus to a small Corliss engine.

When the air supply to the engine was shut off by closing the throttle valve and the compressor allowed to run, wasting air through the relief valve, it seemed upon first thought that the manometer should stand practically at zero. But upon taking into consideration the principle on which it acts and the fact that the compressor caused slight pulsations or fluctuations in pressure with each stroke, which were probably accentuated more or less on account of the inherent sluggishness of the weighted type of blowoff valve used in this case, the following explanation seemed to clear up the matter in question.

The reduction in cross-sectional area of the air passage, known as the "throat" of the venturi, causes the air velocity to increase at this point with a corresponding decrease in pressure. After passing the throat, the gradual flare of the device to the same cross-sectional area as before reduces the velocity until the air pressure becomes practically equal to what it was "upstream." This difference of pressure at the throat is a measure of the volume flowing, called the "venturi head."

Referring to the illustration, suppose P_1 is the pressure upstream. The arrow indicates the normal direction of flow. P_2 represents the pressure at the throat, it being less than P_1 by an amount h shown by the manometer. The pressure at P_3 after the air has passed through the meter is greater than P_2 at the throat, and may be assumed to be practically equal to P_1. If the direction of flow is reversed, the pressures P_3 and P_1 will both be greater than the throat pressure P_2, as before, hence the manometer will read in the same direction.

It is apparent that every impulse of the compressor is accompanied by a slight increase in pressure throughout



METER CONNECTED BETWEEN TWO RECEIVERS

the system, immediately followed by an equivalent decrease due to escapement through the blowoff valve. This continuous rise and fall of pressure, however slight, produces a surging of air back and forth through the venturi throat, and any movement of air at that point causes a difference in level of the liquid in the manometer, regardless of the direction of air flow through the meter. Bearing in mind that the density of air is much less than that of water (the liquid in the manometer), it is evident that the movement of air decelerates and reverses while the greater inertia of the water column keeps the height practically constant, all of which explains the peculiarity of a venturi meter indicating flow when no air was being used and also points out the possibilities of error due to pulsations or rapid fluctuations of pressure where such an instrument is used.

L. A. Wilson.

Urbana, Ill.

Comparative Quality of Oils

In reply to the letter of G. A. Smith, on the "Comparative Quality of Oils," in the issue of June 13, page 848, I would like to offer the following criticisms:

The suitability of a lubricating oil is determined entirely by the conditions governing its use. An oil must be chosen, according to its physical and chemical properties, to fit the peculiarities and difficulties met with in each particular case. It therefore seems illogical to adopt an arbitrary mathematical rule, without a mathematical basis of reasoning, to fit a so widely varying series of conditions as found in efficient lubrication.

An analysis of the rule given by Mr. Smith will show how much it falls short of being a reliable indicator. In this rule the flash point is the controlling factor, while in practice it is one of the least influential of the physical properties except in steam-cylinder lubrication. The viscosity, which is of the greatest importance, has a very slight influence on the percentage quality as determined by the rule given. A change in viscosity of 100 seconds Saybolt (about 2.8 Engler degrees) at 100 deg. F. will cause quite a difference in the frictional loss, but in the quality determination shows practically none. The cold point has little influence on the lubricating properties of an oil used on bearings operating at ordinary temperatures. The suitability is not dependent on specific gravity, but is only a means of identifying the crude from which the oil is derived. Free acid is almost always absent in straight-run pure mineral lubricating oils, while the carbon content as generally determined is so small in these oils as to be negligible.

An examination of the table of results on four very acceptable oils given herewith will show the incompatibility of the rule under discussion. It will be noticed that the percentage quality is almost wholy dependent on the flash point, which is an entirely wrong basis for judgment.

	Oil No. 1	Oil No. 2	Oil No. 3	Oil No. 4
Viscosity, 120 deg. x 1	4.9	5.0	3.1	3.5
Viscosity, 180 deg. x 10	20.5	20.0	18.0	18.0
Flash point	338.0	455.0	334.0	428.0
(1) Sum	363.4	480.0	355.1	449.5
Cold point	0	plus 20	minus 5	plus 25
Free acid	0	0	0	0
Carbon residue	0	0	0	0
(2) sum	0	plus 20	minus 5	plus 25
Difference of (1) and (2)	363.4	460.0	360.1	424.5
Difference ÷ 5	72.7	92.0	72.0	84.9
Specific gravity	0.937	0.885	0.910	0.878
Per cent. quality	68.1	81.4	65.5	74.5

From the table we see that oil No. 2 rates higher than No. 1 and is therefore classed, according to this method, as a better oil. If we add 0.5 per cent. of free acid to No. 2, the percentage quality will be $\frac{480 - (20 + 50)}{5} \times 0.885 = 72.6$, which is still above that of No. 1, and yet, because it contains free acid, it is not at all acceptable.

The mention made by Mr. Smith of a variation in the formula for different classes of oils permits of a change in only the constants used. All four oils, in the table, intended for forced-feed lubrication purposes, have been fully analyzed and tested under operating conditions and found satisfactory. Considering this one class of oils a variation of 13 per cent. quality is found, for different oils submitted for the same purpose, having the same viscosity but from different crudes. Oil has been examined and found to have practically the same physical constants as those used in finding the quality of No. 3 in the table, but it was not suitable for the same purpose.

All this shows the inconsistency and impracticability of attempting to combine the chemical and physical characteristics of a lubricating oil to form an empirical mathematical rule for the determination of quality.

Annapolis, Md. H. R. Smith.

Marine' Engineers Monument at Liverpool

In going over the Aug. 22 issue of *Power,* I noticed, with much interest, your article under the heading of "Monument at Liverpool in Honor of Marine Engineers."

As I hold both a British Board of Trade certificate and a certificate of the United States for chief engineer in unlimited-size steamers, I can readily appreciate the merits of this article and felt that I could not let it pass without writing you a few lines.

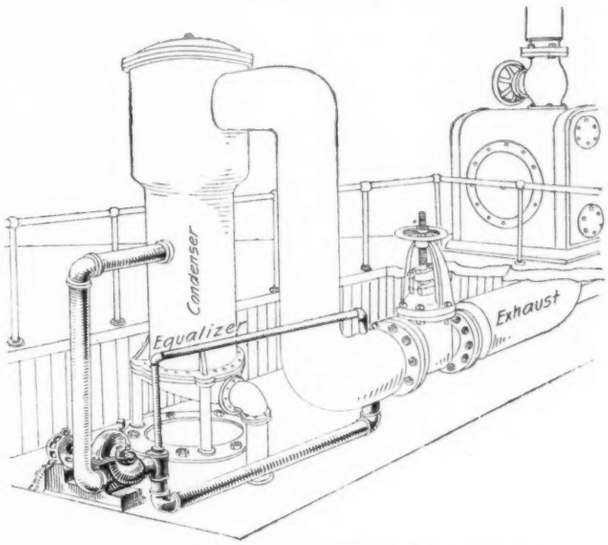
I want to congratulate *Power* for publishing this article in honor of the profession of marine engineering, of which I can impartially say I have never read its equal. I

am sure any marine engineer reading this article can but feel grateful to you for the manner in which you have so clearly brought this profession to the eyes of the public.

Northampton, Penn. H. T. RAISBECK.

Pumping from Vacuum Pipe

The arrangement of the vacuum exhaust piping in a pit made it impossible to locate the draining pump below the pipe so as to get a head of water on it. Accordingly



SUCTION-PIPE EQUALIZER CONNECTION

it was set, as shown in the illustration, with the suction on a level with the bottom of the elbow. It was therefore difficult to get the pump to start.

After several experiments I succeeded in equalizing the pressure or vacuum by placing a tee in the suction pipe and connecting a pipe from this to the upper side of the exhaust pipe. This arrangement has given perfect satisfaction ever since. The pump discharges into the jet condenser above the water level, so the vacuum is practically equal on the suction and the discharge of the pump. It is only a question of priming the pump when starting up. J. S. SHEFFIELD.

Hattiesburg, Miss.

Efficiency of Rope Drives

To one who has not had experience with rope drives it may be difficult to understand why a single strand of rope will show a higher efficiency than two or more. The natural inference is that if one rope has an efficiency of 97 per cent., a drive of ten should have the same.

To have each individual rope pull exactly its share of the load each groove must be exactly the same pitch-diameter and shape. Furthermore, the ropes themselves must be of the same diameter and tension when running, so that each will rest at the same depth in the groove. If this condition could be realized, then the same efficiency could be attained with a number of ropes side by side as with one, but this degree of perfection is never realized hence the actual efficiency depends largely if not altogether on the degree of accuracy in the sheaves and uniformity in the size and tension of the ropes. No matter how small the difference in the pitch diameter may be, the rope or the groove having the greatest diameter in the driving sheave, or the one having the smallest in the driven sheave, will pull the hardest. Assuming this difference to be 0.01 in., it will be 0.0314 in the circumference and in 100 revolutions the rope in one of these grooves would have drawn ahead of the others more than 3 in. and taken more than its share of the load; therefore, this rope or some other must slip. In most rope drives some of the ropes are slack on the pulling side and they can be heard creeping in the grooves. This accounts for the relatively low efficiency of the rope drive. WILLIAM HIRST.

Trenton, N. J.

Gag-Pot Check Valve

A simple Corliss engine directly connected to an alternator the voltage of which was regulated by a voltage regulator ran all right for about three years, when the load had increased until it hovered around 50 per cent. overload. An electric welding machine on the line made things worse, as it acted like a short-circuit every time it was applied. A duplicate of the first engine was installed, and as the engineer had had no experience with alternators, his troubles began. If he did not get them synchronized, they would surge continually; first one would take the load, then the other. He said the trouble was in the new generator. He soon resigned to take up a different line of business, and another engineer was employed. When the new man saw how the machine acted, he first changed the pick-up blocks on the older engine, the edges of which were worn badly and would not hook on. He then removed the

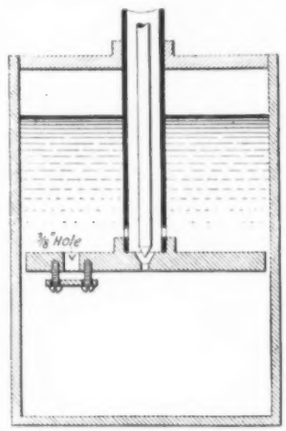


FIG. 1. CHECK VALVE ON PISTON

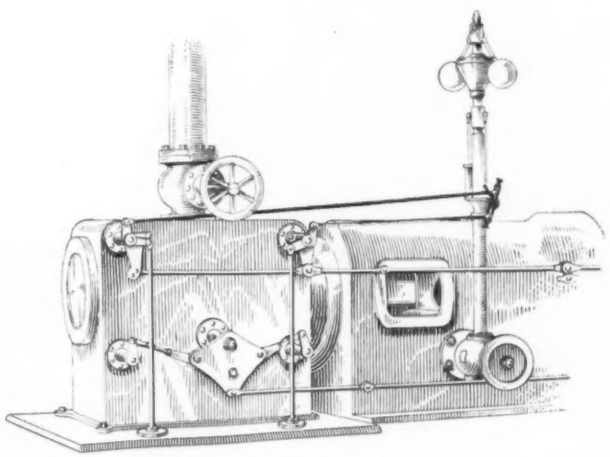


FIG. 2. ADJUSTMENT FOR KNOCKOFF CAMS

pistons of the governor's gag-pots, and put check valves on, as shown in Fig. 1. These consisted simply of a ⅜-in. hole drilled through each piston and a piece of brass placed underneath and held in place by two small screws, the holes in the brass check being large enough to allow it to slide up and down on the screws.

When the governor descended, the check valve would close and prevent the engine from taking on excessive load. If the load dropped off suddenly, the check valve would open and allow the governor to ascend freely. The downward movement can be regulated by the needle valve in the center.

When the machines were operated again, they ran together nicely, but No. 1 machine would take more than half the load when extra load was put on and less than half when the load dropped off. This was corrected in the rods leading from the governor arm to the knock-off cams as shown in Fig. 2. A screw moves the rod block connection up or down to give the cams more or less throw. In this case No. 1 engine was given less throw to make it take half the load no matter what load was on the machines. THOMAS SHEEHAN.

West Springfield, Mass.

Water Chamber for Injector

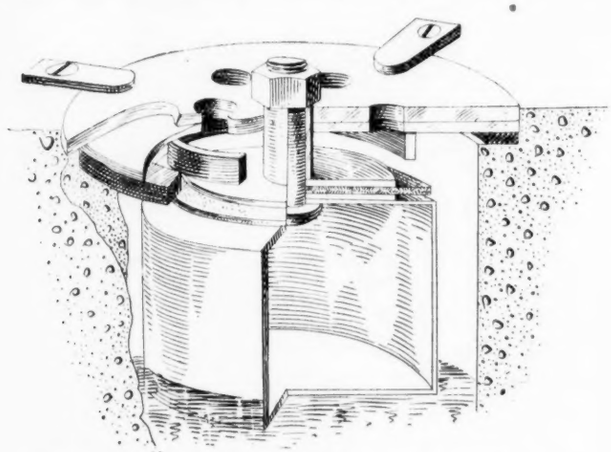
The water supply for our plant comes some distance through a small pipe from the main. After changing from the well to a direct connection to the city supply, the injector did not work satisfactorily for a while.

I made a water chamber from a 4-ft. length of 2-in. pipe, using a reducing coupling on each end to match the suction of the injector. This I placed in the feed-water supply line as near the injector as possible, and the injector has worked well ever since.

Laurel, Md. STANLEY RADCLIFFE.

Check Valve for Sewer Trap

To prevent sewage backing into the basement through a drain having a perforated cover, a check valve was made for it as follows: After the hinge-pins had been sawed off, the cover was taken up and faced off in a lathe



FLOAT VALVE TO PREVENT SEWAGE BACK FLOW

and a circular brass sheet $\frac{1}{8}$ in. thick and 5 in. diameter was sweated to the cover to stop up the holes. Then eight new $\frac{1}{4}$-in. holes were drilled through it. A seat for a rawhide disk was made extending $\frac{1}{4}$ in. below the lower face of the cover and sweated to the brass sheet fastened to the cover. This made a raised seat that is better than the under side of the cover would be. A float was made of copper tubing $\frac{1}{64}$ in. thick and 4 in. diameter and $2\frac{1}{2}$ in. long and soldered to a $\frac{1}{4}$-in. brass rod that works in a brass sleeve $\frac{1}{2}$ in. long in the cover. A circular piece of rawhide $2\frac{1}{2}$ in. diameter and $\frac{1}{16}$ in. thick was placed between two brass disks, 2 in. diameter and $\frac{1}{16}$ in. thick. A $\frac{1}{4}$-in. rubber gasket was placed in the recess under the cover, which is secured to the floor by four $\frac{1}{8}$-in. straps 1 in. wide. This improvised check valve has remedied a great nuisance caused by sewage backing up during heavy rains. W. T. OSBORN.

Newark, N. J.

The Care of Leather Belts

I have read with interest Mr. Benefiel's article on the care of leather belts in the issue of July 25, page 124, and would like to add something to it.

For years it was the custom to have the laps reinforced by rivets, and some buyers still specify that belting shall be riveted at the laps. It is noticeable that riveted belts always show first signs of wear at the rivets, and an investigation discloses two conditions: First, that whenever so constructed, the belt is less flexible at the lap than elsewhere and the more rigid lap strikes the pulley a blow, with the result that that particular portion of the belt slips and burns; second, that the rivets, being harder than the leather, are pounded in and cut it.

The abandonment of the use of rivets in leather belting has become quite general, and the belt maker is now applying his talents to producing a lap that is as near the consistency of the other parts of the belt as possible. As a result, it is impossible to locate the laps by the sense of touch and such a flexible lap eliminates the blow and insures the greatest durability.

A properly laced belt, where a flexible lace is employed, is frequently preferable to a lap that is too rigid. The lace takes the force of the blow and, being softer than the belt, does not injure it; and the occasional replacement of the lace is more economical and practical than piecing a belt that has been worn at a too rigid lap.

Care must be exercised as to the thickness in selecting the lace. A lace that is too thick will not only wear itself out, but will cause a loss of power by breaking the contact between the belt and pulley.

Philadelphia, Penn. GEORGE H. MORGAN.

What Blisters the Boilers?

The cause of deposits and blistering at the one particular spot, 12 in. from the front heads, as described by George F. Wright in the issue of Aug. 1, page 173, is no doubt excessive heat at this point, which forces rapid circulation of the water upward, leaving the sediment there. The extra heat is probably due to the grates being imperfectly covered directly inside of the brick pier between the fire-doors, possibly owing to extra width or length of this pier, making it difficult to properly cover the grates at this place and allowing more air to enter, causing a blowpipe effect on the bottom of the shell.

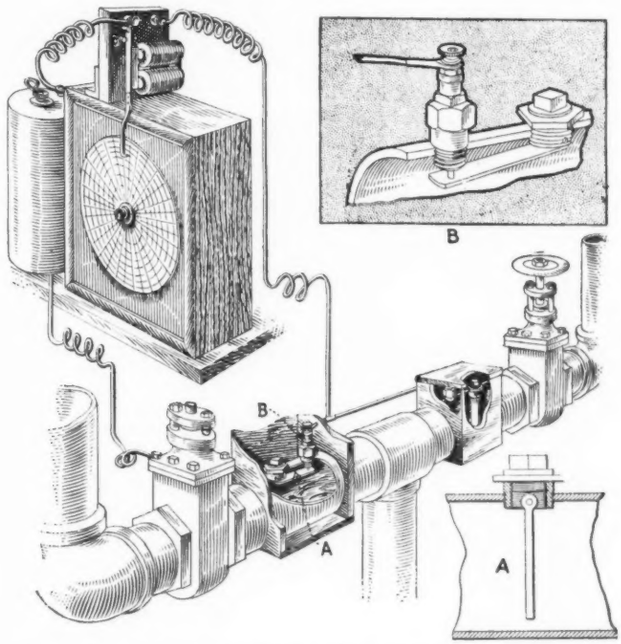
This condition may be overcome by inserting a 2-in. pipe in the boiler front and through the brick arch, directly under the center and as near the boiler shell as possible, to admit sufficient cool air to prevent burning the plates. SOREN ANDERSON.

Butte, Mont.

Heavy Mineral Oil with an addition of heavy animal or vegetable oil is best for high-speed shafts and electric machines having considerable weight to carry.

Record Time of Blowing Boilers

We have felt the need of some means of recording when the blowoff valves of our boilers were opened to blow out mud and sediment. The following system is in satisfactory use on five boilers: In the blowoff pipes of each boiler is a brass hinge *A*, which, when water is passed by opening the blowoff valve, raises the hinge, causing it to make contact with the spark plug *B* screwed into the pipe. The



HOME-MADE DEVICES FOR RECORDING WHEN BOILERS ARE BLOWN DOWN

hinge and plug are wired to buzzer magnets having a steel-wire spring for punching the chart of a watchman's clock. Current is furnished by a battery that forms part of our firebell ringing system. Moreover, the relative positions of the spark plugs and hinges are such that contact for a particular boiler is made only if that boiler is blown.

WILLIAM E. ADAMS.

Philadelphia, Penn.

Phantom Warnings

W. O. Rogers' "Phantom Warning" stories are interesting as well as novel and bring to mind some narrow escapes by some unforeseen incident drawing a person away from the danger zone.

It is the rule in our plant to wipe up the engines soon after they are shut down. After wiping the frame, valve gearing, etc., of a cross-compound engine shut down a few hours previously, I climbed into the flywheel to wipe the inside of it. While in there I thought I heard something about the running of the other engine that did not sound right. I hesitated about climbing out of the flywheel on a wild-goose errand, as I saw nothing unusual, I had no premonition and no "Phantom Lady" tried to lead me out, but anyway, I crawled from the wheel. I had no more than stepped out when the wheel started to revolve as the engine could be heard getting steam.

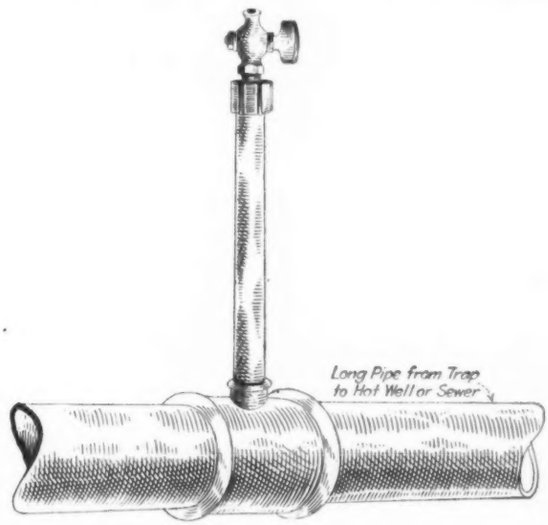
On investigation the mystery was cleared up. The bypass was not closed tight, and the steam pressure was high. The following is the only way that I could account for the fact that the engine had not started before. After the heavy-load period the steam pressure was dropped from 155 lb. to 130 lb. and one engine was shut down. For some reason the fireman allowed the pressure to go above 130 lb., and at the higher pressure the steam accumulated in a sufficient quantity to start the engine. What about the running of the other engine? I could not find anything wrong, but the something or nothing had saved my life because I could not have got out of the wheel while it was in motion.

THOMAS M. GRAY.

Middletown, N. Y.

Telltale for Steam Traps

Where there is a long run of pipe from a steam trap to a hotwell or sewer, there may be considerable undetected leakage of steam from the trap. A try-cock on a telltale pipe, as shown, will often show up the trouble.



TRYCOCK ON TRAP DISCHARGE

If only hot water is leaving the trap a light vapor will escape, but if considerable steam is escaping it will show quite a blow. This simple attachment will often prevent a big waste.

JAMES E. NOBLE.

Toronto, Ont., Canada.

CO_2 Recorder in Small Plants

In reply to Mr. Priefer, Aug. 1, page 178, I believe an expensive CO_2 recorder is not a good investment for small installations, especially since each boiler should have a recorder and a considerable cost must be figured in labor, maintenance, interest and depreciation; and intelligent handling is necessary if the instruments are to be kept in reliable working order.

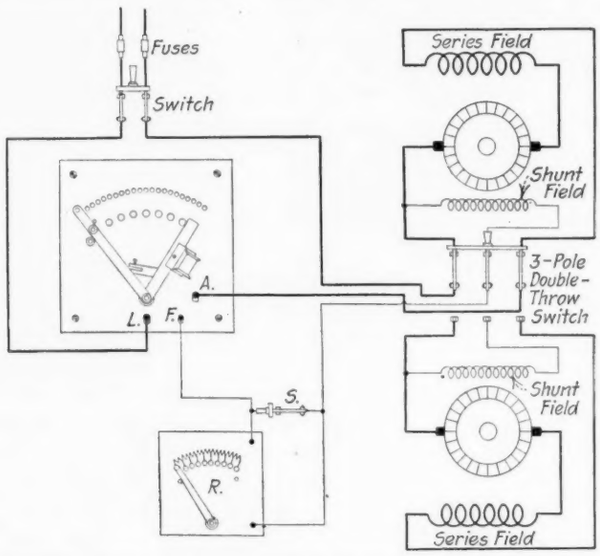
The recorders should be placed in the boiler room, as the lag—that is, the time between taking the sample and the record indicated upon the chart—depends on the distance between the boiler and the recorder.

In small plants the best combination is the Orsat and collecting bottles. Analyze the sample collected at the end of each run and build up with the Orsat. All collecting bottles should be equipped with a float and needle valve to give a constant rate of flow regardless of head.

Middletown, N. Y. HARRY A. COX.

Emergency Motor Connections

The illustration shows a system of connections which was devised to provide emergency service with two motors and one regulator. The three-pole, double-throw switch between the two motors makes it possible to start either



CONNECTIONS FOR STARTING TWO MOTORS FROM ONE STRATING BOX

motor at will. The field resistance of the regulator not increasing the motor speed sufficiently, an additional resistance *R* was inserted in the field circuit, as shown. As the resistance was not adjustable, a single-pole switch *S* was connected across the resistance terminals so that it could be cut in or out of the field circuit by opening or closing the switch. This arrangement, while providing emergency service if one of the motors fails, would not, if the regulator failed. R. L. HERVEY.

Washington, D. C.

[Temporary service may be secured in the event of regulator failure by using a water rheostat.—Editor.]

Efficiency of Unsymmetrical Riveted Joints

There are certain statements that are open to question in R. N. Blackburn's article on the "Efficiency of Unsymmetrical Riveted Joints," in the issue of Aug. 8, page 203.

The calculation on the top of page 203, second column, would be very nearly correct if both shell and rivets were absolutely rigid, but such is not the case and a slight deformation within the elastic limit would cause the second row of rivets to take on a greater load. I believe that the efficiency is nearer 70 than 52.5 per cent.

In reference to the influence of strap thickness, I think that if cover plates are designed with scalloped edges, as shown in my article in the issue of Aug. 1, pages 159 to 162, the stresses in the net section will be nearly the same for both shell and strap so that the deformations of both will be almost alike and should cause no serious unbalancing in the distribution of load among the rivets.

The illustration of the elastic rubber, although interesting, may cause incorrect conclusions. If, for instance, the shearing rigidity were negligibly small as in the case with rubber, the fibers in the immediate vicinity would be stressed just as he has pointed out, but since the shear between fibers in boiler plates is large, no very great deformation of fibers in the immediate vicinity of the rivet can take place. If this is carried to the extreme—that is, perfect rigidity assumed—the stress over the section between rivets will be uniform. As steel plate is more nearly rigid than rubber, the conclusion in the case of rubber is likely to be divergent from the facts.

Mr. Blackburn is not clear in that part under the heading, "Stress at Right Angles to Pull," and I would like to have more information as to the conclusions drawn, forming a law of variation of stress.

So far as criticism of the A. S. M. E. Boiler Code is concerned, I agree with Mr. Blackburn. It is unfortunate that these joints are sent out leaving others to suppose them standard. Subsequent editions will perhaps see a change in this part of the report.

The report entailed a large amount of work and is admirable as a step in the right direction.

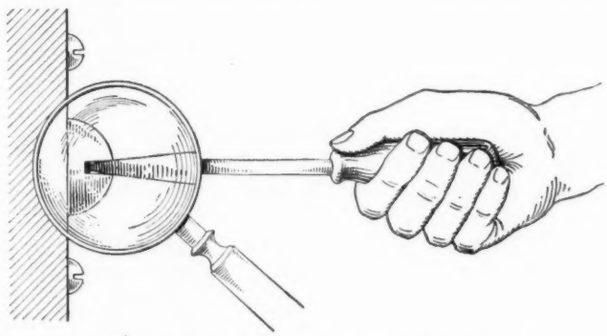
I believe that the method given by me in the issue of Aug. 1 is on as secure a foundation as the subject of "Strength of Materials." For example, if a bar breaks at 60,000 lb. per sq.in. and in subsequent design the area is calculated from this test datum, whatever mistake is made in ascertaining whether the true unit stress is 60,000 lb. matters little, since this same error is reintroduced in the reverse form when determining the required area for a given load, so that the effect is one of counterbalancing errors. If this is done with riveted joints to determine the various ultimate stresses from actual tests, the simple methods of calculation would give sufficiently accurate results.

There is no present need for a more refined theory of riveted-joint design, since the ultimate strength of the boiler plate varies, in some cases over 10 per cent. in successive strips along the shell and the theory advanced checks with a smaller error than this. A. A. ADLER.

Brooklyn, N. Y.

Correct Shape for Screwdriver

The illustration shows how to file or grind a screwdriver so it will tend to hold itself in the slot of a screw and not slip out. The sides of a screwdriver should be



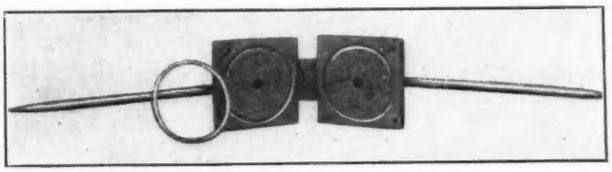
WELL FORMED END ON A SCREWDRIVER

flat and parallel to fit the slot in the screw and not, as usually found, beveled like the point of a chisel. For especially hard jobs the concave form shown is advantageous. E. M. KEYS, JR.

Montesano, Wash.

Mold for Lead Gaskets

The handhole gasket mold shown in the illustration is made of two steel plates 6x6x¾ in., each having a circular groove 4 13/16 in. inside diameter and about ⅛ in. deep, making a gasket ¼ in. thick. The plates are held



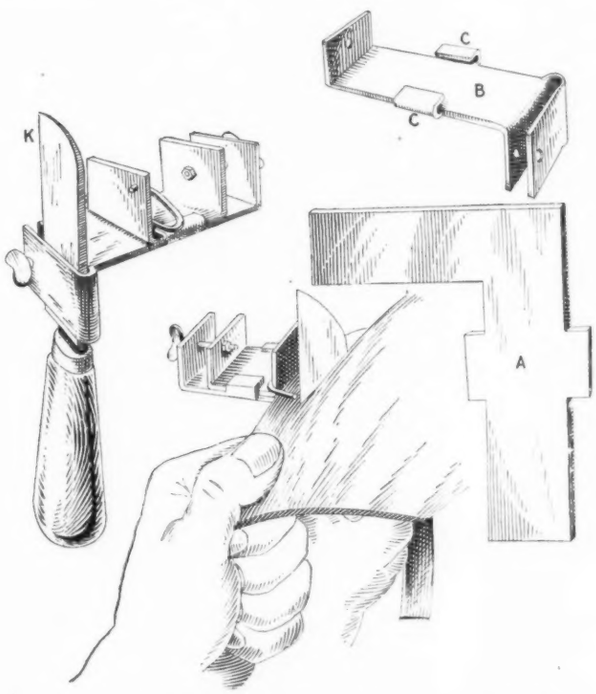
GASKET MOLD AND ONE OF THE RINGS

together by an ordinary gate hinge, and two dowel pins bring the grooves into position when closed with the handles to be seen extending from the back of each plate. The gaskets as made are round, but can easily be shaped to the elliptical form of the handhole plate.

Alicia, Penn. F. G. RUSKIN.

Gage for Splitting Belt Lace

A simple device for a gage for splitting belt lace is made of heavy sheet brass or iron into the shape shown at *A* and bent as shown at *B*. The short extension *C* is turned over to form a groove for the slide, as shown assembled. The adjusting screw holds the slide in posi-



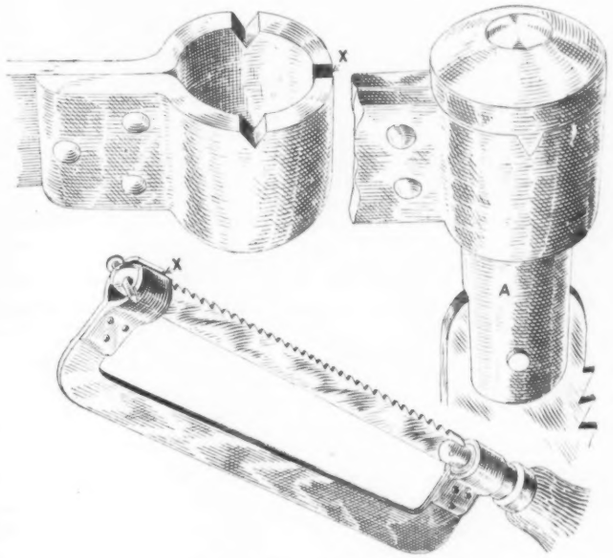
GAGE TO CUT BELT LACE TO UNIFORM WIDTH

tion for the width of lace required. A small spring wire is bent and fastened to the slide with the end extending over the lace, as shown, and keeps the lace level on the gage. This little contrivance is placed on the blade of an ordinary jack-knife *K* and clamped. The whole thing is only 1¾ in. long and ½ in. wide. It is easily made and serves the purpose as well as an expensive gage, which is seldom provided.

Toronto, Ont. J. O. MCDONNELL.

Fitting a Saw Frame to Blade

I had urgent occasion to use a hacksaw, but found that I had only a 10-in. solid frame and 12-in. blades. There was no possible chance to get an adjustable frame or 10-in. blades, but I was able to use the long blades by remov-



LONG BLADE IN SAW FRAME

ing pin *A* entirely and sawing a slot at *X*. When the 12-in. blades run out of stock I can again use 10-in. lengths by inserting the original pin in its place.

New York City. H. E. HOFFSTADT.

Making Good on Reports

The responsibilities of the power-plant engineer increase his opportunities for usefulness. If he shows fitness for advisory tasks, the opportunities to carry out special investigations and the manner in which such duties are handled often influence his future. The preparation of reports deserves special attention. Routine duties are sometimes allowed to absorb too much of the time that should be devoted to investigation and valuable reports.

A case in point is one where an engineer, under instruction from his employer, left the charge of a plant to his assistant and devoted six weeks' time to the gathering of information from various cities and factories, the knowledge gained to be embodied in a comprehensive report. He read everything he could lay his hands on that had a bearing upon the subject under investigation and sent a tabular list of questions to other concerns in the field accompanied by a letter explaining why the information was wanted. He assembled the replies in manila envelopes, keeping the different data separated, and listed the sources of all information. Published statistics helped in many cases, and in some instances public utility commissions' reports threw light upon the problem. Prompt acknowledgment was made of data or facts supplied by others.

When the task was well in hand, he devoted several days to computations of unit cost, etc., inscribing all data on tracing cloth in ink—a plan that enabled late news to be added to the report, even on the final

day. Finally, blueprints costing but a few cents each were made, to be presented with the finished report. The engineer kept a carbon copy of every sheet and an extra blueprint of each item for his own file, making it possible to answer subsequent questions promptly.

When the task was completed, it contained information sheets in alphabetical subject order and complete to the last detail within the limits of the available time. A page or two of comment or synopsis came first. The body of the report contained matter that could at once be located. The limits of accuracy embodied in the figures given were stated, and throughout the report the authority for every statement not made by the author himself was noted.

The report received due appreciation, and the engineer returned to routine duties glad that he had been afforded the privilege of making the investigation, which was as good as a post-graduate course in the special field covered.

The importance in such work of clear-cut understanding of exactly what the problem is, of seeing to it that every computation and statement can be verified or ascribed to due authority, of keeping reports free from superfluous matter, and of presenting them in useful form need not be repeated. H. S. KNOWLTON.

Cambridge, Mass.

The Analysis of Riveted Boiler Joints

In answer to the criticism of my article on "Graphic Analysis of Riveted Boiler Joints" (Aug. 1 issue) in the issue of Aug. 22, I am sorry to note that the point is entirely missed in Mr. Jeter's discussion. The analysis presented by me was given to answer a question like this: Given a definite value for resistance in tension, shear and crushing, what is the best riveted joint that can be made with a given number of rows of rivets? There is only one answer, and that I have given in the ideal case. If there is more than one answer, then either the theory is indefinite or the question is improperly put.

Mr. Jeter's criticism is, however, on special cases arising and in many places is indefinite. For instance, in paragraph four there appears: "The next commercial size is ¾ in. but $\frac{15}{16}$ in. or 1 in. may be just as well used . . . ." Why stop at 1 in.? If you can go as far as you like, then there is no limit. Continuing in paragraph five, we meet the statement that, "While the design of a joint of maximum efficiency is of interest, what is of more importance to the designer is to be able to tell the proper pitch to choose for an arrangement of rivets that has proved to be best suited to the requirements of boiler construction, . . ." I see here three questions: If you are at liberty to choose many diameters of rivets, what is the proper pitch? When is the best-suited requirement obtained? Who is the judge? To me, at least, statements like those quoted convey no meaning. I intuitively substitute the words, "I don't know," for all these indefinite assertions, although I do not mean to imply that this is Mr. Jeter's case. It is the purport of a theory to answer just such questions rather than span the domain from anything to everything.

In paragraph two, Mr. Jeter takes a special case and finds that if I had $T = 50{,}000$ lb. instead of 60,000 lb. a suitable ratio of rivet spacing from row to row would result. Now this is juggling, not engineering. If I am permitted to assign any values to T, S or C, then I can make a joint of any assignable efficiency no matter how high, with a single row of rivets. This is not a remarkable statement; but it is absurd. We are concerned with specific properties of materials; that is, their resistance to particular forces, and these we must accept as experimentally determined. Given these values, we get one and only one answer in riveted-joint design if the theory is consistent. That there may be other joints nearly as good is another matter, but we must settle the question for the ideal limit and then determine the practicability from the economics involved. As I presented it, the theory is applicable to any plates connected by any kind of rivets provided we know the required properties of the materials.

In paragraph three Mr. Jeter comes to the conclusion as to what I do not understand about riveted joints and then proceeds at the beginning of paragraph five to tell us that if you choose larger rivets you must increase the pitch for higher efficiencies. We then read at another place about "calkability." Mr. Jeter knows that this is best obtained with the smaller pitches.

One more point upon which I wish to take issue with Mr. Jeter is in the calculation of strap thickness. At the very outset we assume that the stress is equally divided among the rivets, a statement open to question. The more rigid we make the strap the less will be the stress carried by the rows of rivets near the center line of the joint. I believe that the strap should be designed so as to deform as nearly as possible the same amount as the shell. Then if the stress is evenly distributed among the rivets at the start, it will be more nearly in accord with the original assumption after the load is applied.

As I review the analytical treatment, I again come to the conclusion that this is by far the most satisfactory, although it is contrary to Mr. Jeter's opinion. I do not think one designs a joint when he chooses an arrangement from another's chart. That is copying and not designing. The subject of riveted joints is far too simple to cloud with mysteries supposedly understood only by the initiated.

On the whole, I thank Mr. Jeter for reviewing my article. ALPHONSE A. ADLER.

Brooklyn, N. Y.

Seamless-Drawn Tubes and Pipes are made from a solid round billet drilled in the center at one end, then heated to a soft plastic state and pierced. The rough tube is then passed between grooved rolls with mandrels of the proper size inside. This process produces finished tubing and pipe of remarkably uniform size and quality. For sizes over 5½-in. the so-called "cupping" process is used, in which circular plates of best-quality steel are heated to a bright-red and pressed first into a cup shape, reheated and the process repeated, using smaller dies, until the metal is in the form of a tube with one closed end. Then it is forced through dies on the drawing bench until of the proper caliber and gage. Frequent reheatings are necessary during the process. When cut to length by removing the irregular ends, the pipe is ready for testing.

Material Used in "Shelby" Tubes of the three principal classes contain 0.17-per cent. carbon steel, 0.35-per cent. carbon steel and 3½-per cent. nickel steel. The first-named quality is suitable for boiler tubes and purposes requiring ductility, the second where higher elastic limits and ultimate strength are required, and the last where ductility is necessary together with high elastic limits and ultimate strength. Stock piping of a great variety of chemical compositions and sizes to meet almost any requirements may be procured without resort to special orders.

Inquiries of General Interest

Pump-and-Receiver Trouble—Trouble is experienced in the use of a pump and receiver from racing and slamming of the pump whenever the principal returns of a heating system are cut off. What are the probable cause and remedy? C. R.

Racing and slamming would occur if the float-controlled throttle valve of the pump is open or leaks when there is not enough water in the receiver to supply the pump. The remedy would be to repair the pump throttle so it will be tight when closed and adjust the connections to the throttle float so the float will close the throttle for a higher level of water in the receiver.

Thermal Efficiencies of Large Power Plants—What is meant by the thermal efficiency of a plant, and what is considered to be the average thermal efficiency of power plants? L. T. S.

The thermal efficiency of a plant is the ratio of the heat equivalent of the power output to the heat input, or heat contained in the coal or other fuel that is used. Some of the largest condensing steam-turbine plants have realized thermal efficiencies as high as 17 per cent., and the average thermal efficiency of large turbine plants is probably about 10 per cent. while that of reciprocating-engine plants of 750 to 1,500 hp. is 8 to 9 per cent.

Bolt Loose in Engine Cylinder—How could a ¾x4-in. bolt find its way into the cylinder of a 16x36-in. engine? The engine had been running for about eight hours at regular speed, 100 r.p.m., when suddenly there was a violent pounding and rattling in the cylinder and the engine stopped. The piston, otherwise intact, was found to be wedged fast by the bolt. F. B. E.

It is probable that the bolt was dropped into the steam passages of the cylinder before the steam pipe was connected to the engine and possibly as early as when the cylinder was machined, finally finding its way to the inside of the cylinder just as pieces of foundry cores, sand and scales of cast iron may become dislodged and swept into the cylinder some time after an engine has been in use.

Expansion and Contraction of Lead Pipe—Does lead pipe expand and contract with changes of temperature? J. J. W.

At ordinary temperatures the linear expansion of lead for each degree Fahrenheit increase of temperature is about 0.0000157 of its length, and the coefficient of contraction per degree fall of temperature is substantially the same. That lead pipe does expand for an increase of its temperature is readily seen from the increase of length in the bends that occur between points of fastening of a straight run of lead pipe used for hot water. When the deformation from expansion has resulted in stresses of the material beyond its elastic limit, a change has occurred in the arrangement of its particles and a return to the lower temperature will not entirely restore the original alignment, hence repeated heating and cooling a lead water pipe may eventually result in considerable change in its alignment.

Saving from Increasing Temperature of Feed Water—In the generation of steam at 90 lb. boiler pressure and using an exhaust-steam feed-water heater without increasing the engine back pressure, what saving of fuel would be effected by raising the temperature of the feed water to 200 deg. F. in place of 110 deg. F.? W. L. J.

The absolute pressure at 90 lb. boiler pressure would be about 90 + 15 = 105 lb. absolute, and by referring to the steam tables it may be seen that the total heat of a pound (weight) of dry saturated steam at that presure is 1,187.2 B.t.u. above 32 deg. F. With the feed water at a temperature of 200 deg. F., or 200 — 32 = 168 deg. F. above 32 deg. F., each pound of feed water for conversion into dry saturated steam at 90 lb. boiler pressure would have to receive 1,187.2 — 168 = 1,019.2 B.t.u.; but for conversion from feed water at a temperature of 110 deg. F., or 110 — 32 = 78 deg. above 32 deg. F., each pound must receive 1,187.2 — 78 = 1,109.2 B.t.u. Therefore the saving would be $\frac{1,109.2 - 1,019.2}{1,109.2} = 0.081$, or about 8 per cent.

Transmission of High-Voltage Direct Current—Has high-voltage direct current been used for long-distance electric transmission, and if so, what is the highest voltage used? Has this method any advantage over the alternating-current system? P. M.

The well-known Thury system of direct-current transmission used in Europe employs voltages up to 100,000. This is a constant-current system, the high voltage being obtained by connecting series direct-current generators in series. The highest voltage taken from one commutator is 5,000 volts.

The advantages over an alternating-current system are the absence of inductance, capacity, and corona; high-tension underground cables can be used, single-conductor cables having been used up to 100,000 volts. The disadvantages are that this system is only a transmission system and not a distributing system. Before the current can be distributed, it must be converted into either low-voltage direct current or alternating current through a motor-generator set. All machines must be insulated from ground, and the high-voltage commutators are a source of trouble and expensive to maintain.

Power Lost from Friction of Shafting—What is the formula for estimating the power absorbed by the friction of shafting? H. J. S.

The power absorbed by the friction of horizontal shafting running on cylindrical journals in babbitted bearings, based on the average of frictional coefficients established by tests of M. Morin (France) and those derived from results of extensive observations of the resistance of mill shafting by S. Webber (United States), is given for each bearing by the formulas,

$$\text{Hp.} = \frac{W \times d \times \text{r.p.m.}}{1,800,000} \text{ or about } (W \times d \times \text{r.p.m.}) \times 0.00000056$$

for ordinary oiling, and

$$\text{Hp.} = \frac{W \times d \times \text{r.p.m.}}{2,950,000} \text{ or about } (W \times d \times \text{r.p.m.}) \times 0.00000034$$

for continuous oiling

in which

Hp. = Horsepower absorbed by journel friction;
W = Total weight of shafting and pulleys, plus the resultant stress of belts, in pounds;
d = Diameter of journal in inches;
r.p.m. = Number of revolutions per minute.

Height of Pumping Water—To what height can water be raised with a direct-acting steam pump supplied with steam at 90 lb. pressure per square inch if the steam cylinder is 8 in. diameter, water cylinder 16 in. diameter and the suction water is supplied at a pressure of 40 lb. per sq.in.? J. T.

The pressure exerted by the water piston will depend on the design, construction and adjustment of the pump, and the height to which a given water-cylinder pressure will raise the water would depend on the pressure required for overcoming the inertia of the water and the amount of pressure lost in overcoming friction. Allowing a drop of 2 lb. in the steam pressure from the throttling action of the steam ports and passages, an exhaust back pressure of 2 lb. and a loss of 6 lb. pressure per square inch for overcoming the friction of piston stuffing-boxes, steam valve and valve gear and friction of pistons, the net steam pressure available for overcoming the resistance to motion of the 8-in. diameter steam piston would be 90 — (2 + 2 + 6) = 80 lb. per sq.in. Without allowance for cross-sectional area of the piston rod, the 16-in. diameter water piston would exert a pressure of $80 \times \frac{8 \times 8}{16 \times 16} = 20$ lb. per sq.in., and allowing 2 lb. per sq.in. loss of pressure for delivery of the suction water to the suction side of the piston, the assistance rendered by the suction-water pressure would be 40 — 2 = 38 lb. per sq.in., making the total pressure exerted by the water piston, 20 + 38 = 58 lb. per sq.in. The pressure due to 1 ft. head of standing water at ordinary temperatures is 0.433 lb. per sq.in., hence 58 lb. per sq.in. would equal 58 ÷ 0.433 = 134 ft. head.

The head lost in discharging the water from the pump cylinder, due to inertia and friction of the water, would depend on the speed of the pump, the percentage of slippage and the size and form of the discharge passages and piping. Under the conditions named and assumed, these losses, for a direct vertical lift, ordinarily would amount to about 10 per cent. of the head due to the actual height, and the actual working lift would be about 122 ft.

Disastrous Boiler Explosion at Jackson, Tenn.

By J. A. Willard

The following is a full report of the boiler explosion that occurred on Monday morning, Aug. 21, referred to in the issue of Sept. 5, page 368, at the plant of the Harlan-Morris Manufacturing Co., at Jackson, Tenn.

The general arrangement of the boiler plant is shown in Fig. 1. As indicated, there were three boilers, of the return-tubular type, supplying steam to two engines (200 hp.) and to a dry kiln. The steam piping consisted of a main header, from which lines to the two engines were taken off and also the line to the kiln. Boilers Nos. 1 and 2 were piped up so that they could feed either directly to the main or to the dry kiln; No. 3 fed to the main only. After the explosion all the header valves and boiler stop valves were found open and their stems bent, indicating that all were in the proper operating position.

The explosion occurred ten minutes after starting time, when all three boilers exploded with great violence. As shown in Fig. 2, the whole factory, with the exception of the finishing mill, was wrecked. Four white men, including the engineer, were instantly killed and the three colored firemen and one helper also met instant death. Ten more were seriously injured, though all of these stand good chances of recovery.

Like most factories of this kind, the walls and roofs were of frame covered with galvanized iron, with the exception of the shaving section, which had brick walls. These were all leveled to the ground. The estimated property damage was about $15,000. The main force of the explosion was directed westward, and boiler tubes were scattered to a distance of 300 ft. in this direction. One curious feature was that a boiler tube cut entirely through more than half of the front of the negro cottage at the left and rear of Fig. 3 and landed on a bed. Another tube made a hole in the roof, but fortunately no one in the house was injured.

Apparently No. 2 boiler let go first. The middle course of this boiler is shown in Fig. 3, and its location when found is indicated at A in Fig. 1. The initial failure appears to have taken place at the steam nozzle, and the tear went through the solid plate, the boiler then opening up in the girth seams. The rear course of this boiler was found standing vertically at B in a mass of timbers. Fig. 4 shows a corner of this sheet, revealing how the ring seam tore out. Part of the back head was with this sheet; the part above the tubes and part of the rear course were found 630 ft. away to the southwest. The front course was found at C and is shown in Fig. 5. At first the lap joint of this course was believed to be the point of initial failure, but a careful examination failed to show any signs of a lap crack. The tear was through sound plate. It is interesting to note in this picture how some of the girth-seam rivets sheared and some pulled through the lap. Fig. 6 shows the other side of this course and also the front head.

Boilers Nos. 1 and 3 were set off by the explosion of No. 2. Two views of the front course and head of No. 1, which was found at E, are shown in Figs. 7 and 8. The middle and rear courses with rear head and steam dome of this boiler were blown 640 ft. away, as indicated at D and shown in Fig. 9. The only failure in this boiler was the shearing of the girth-seam rivets and the pulling out of the tubes.

The front course of No. 3 boiler was found at F and is shown in Fig. 10. The rear course drove straight through to the middle of the finishing mill at G and is shown in Fig. 11. Like No. 1, it sheared all its girth-seam rivets.

The primary cause of this explosion will probably never be known. Either the steam nozzle or a fitting just above it on No. 2 let go first. Finding the boiler stop valve properly opened stopped speculation as to the boilers being connected when the steam pressure was higher in one than in the others. These boilers were under insurance inspection. They were all set separately, in what is known as an all-steel "steamboat" setting. This means that the part back of the bridge-wall was brought in near to the boiler shell, but was elliptical in cross-section. This type of setting gives a restricted combustion chamber; but with it the boiler is hung on steel I-beams, supported by steel columns, and strains in the piping due to settlement are unlikely. There were no signs of low water, and the fact that half of one boiler was thrown 640 ft. shows that there was plenty of energy available. The superintendent of the plant, who was injured but not killed, was in the boiler room two minutes prior to the explosion and reports ample water in the gage-glasses and 85 lb. pressure on the steam gages. The water columns were found with all openings clear, so that the gage-glasses presumably gave a correct indication of the water level. The boiler inspector allowed 90 lb. pressure. The pop safety valve from No. 3 was found in good order, and when tested, it relieved at 85 lb. The manager and several others claim to have heard one of the boilers pop on the Friday before the explosion, so it is unlikely that the initial rupture was due to overpressure within the boiler.

No. 1 boiler was 60 in. by 16 ft. and had fifty-two 4-in. tubes. The shell was $\frac{5}{16}$-in. plate and the heads $\frac{3}{8}$ in. The pitch of the double-riveted lap joint was three pitches to eight inches. It was hard to determine whether the rivet holes were $1\frac{1}{2}$ in. or $\frac{3}{4}$ in., and it makes a difference of less than 0.05 in the factor of safety, which is taken at 4.52. This boiler was 10 years old. It was moved to the present site about two years ago along with No. 2. The plates were in good condition, and it had never been bagged or patched; the tubes showed little scale.

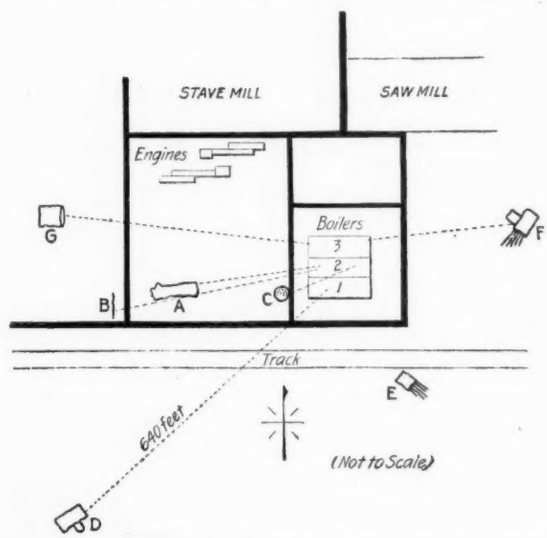


FIG. 1. APPROXIMATE POSITION OF BOILER FRAGMENTS AFTER THE EXPLOSION

No. 2 was about 12 years old, 60 in. by 16 ft., having fifty-two 4-in. tubes. The lap seam was double-riveted and had three pitches in eight inches, with $\frac{7}{8}$-in. holes. Shell plates were $\frac{5}{16}$ in. and heads $\frac{3}{8}$ in. and all were in excellent condition. The factor of safety was 4.27.

No. 3 was a two-course boiler, 60 in. diameter by 14 ft. long with fifty-seven $3\frac{1}{2}$-in. tubes. The sheets were $\frac{5}{16}$ in. and heads $\frac{3}{8}$ in. The double-riveted lap joint had $\frac{3}{4}$-in. rivet holes and $2\frac{1}{4}$-in. pitch. The factor of safety was 4.25. This boiler had been moved to this plant last spring. It was 23 years old, but notwithstanding its age, was sound and the condition of the plate was excellent. Both Nos. 1 and 3 boiler had steam domes, but the peculiar separate dome of No. 2 had been done away with two years ago on the advice of an inspector because it vibrated badly.

This last point might suggest an explanation of the initial failure of the cast-iron steam outlet flange of this boiler. It is possible that the vibration during the previous years of use with the separate dome up to two years ago had caused a fatigue of the metal in the cast-iron outlet flange by which the dome was attached, resulting in its ultimate failure. That this is the true explanation, however, is extremely problematical.

It is anticipated, as a result of this explosion and in view of the large loss of life, that steam-boiler inspection by the state will be urged at the next session of the Tennessee legislature. If so, a determined effort will be made to secure the adoption of the A. S. M. E. Boiler Code as the official code for the state.

FIGS. 2 TO 11 INCLUSIVE. PHOTOGRAPHS TAKEN AFTER THE EXPLOSION OF THREE BOILERS AT JACKSON, TENN.

Figs. 2 and 3—General views of wreckage, cottage in rear (Fig. 3) damaged by boiler tubes. Fig. 4—Part of rear sheet. Fig. 5—Front course of No. 2 boiler. Fig. 6—Same fragment as in Fig. 5. Figs. 7 and 8—Two views of front of No. 1 boiler. Fig. 9—Rear section of No. 1 at D. Fig. 10—Front course of No. 3 at F. Fig. 11—Rear course of No. 3 at G

Sickles Steam-Flow Tables

The table of pipe sizes for a given steam flow and pressure drop per 1,000 ft. at various initial pressures, compiled by E. C. Sickles, is one of the most convenient yet devised. In Table A (diameter and discharge) under pipe sizes from 1 to 12 in. arranged in columns are various quantities (pounds) of steam conveyed by the several sizes of pipe and the loss in pressure for a chosen quantity will be found in Table B (density, pressure and drop) by following over horizontally to the column under the pressure to be dealt with.

FLOW OF STEAM THROUGH PIPES

Table A. Diameter—Discharge										Table B. Density—Pressure—Drop							
1 in.	1½ in.	2 in.	2½ in.	3 in.	4 in.	6 in.	8 in.	10 in.	12 in.	0.208 90	0.230 100	0.284 125	0.328 150	0.401 180	0.443 200	0.506 230	0.548 250
2.52	6.81	18.1	28.8	55.9	123	371	799	1,443	2,328	18.10	16.4	13.3	11.1	9.39	8.50	7.44	6.87
2.34	6.52	16.8	27.6	51.9	114.6	344	742	1,341	2,165	15.60	14.1	11.4	9.60	8.09	7.33	6.41	5.92
2.16	6.24	15.5	26.4	47.9	106	318	685	1,237	1,996	13.3	12.0	9.74	8.18	6.90	6.24	5.47	5.05
1.98	5.95	14.2	25.2	43.9	97.0	292	628	1,134	1,830	11.1	10.0	8.13	6.83	5.76	5.21	4.56	4.21
1.80	5.67	12.9	24.0	39.9	88.2	265	571	1,031	1,663	9.25	8.36	6.78	5.69	4.80	4.34	3.80	3.51
1.71	5.29	12.3	22.8	37.9	83.8	252	542	979	1,580	8.33	7.53	6.10	5.13	4.32	3.91	3.42	3.16
1.62	5.00	11.6	21.6	35.9	79.4	239	514	928	1,497	7.48	6.76	5.48	4.60	3.88	3.51	3.07	2.84
1.53	4.72	10.9	20.4	33.9	75.0	226	485	876	1,414	6.67	6.03	4.88	4.10	3.46	3.13	2.74	2.53
1.44	4.43	10.3	19.2	31.9	70.6	212	457	825	1,331	5.91	5.35	4.33	3.64	3.07	2.78	2.43	2.24
1.35	4.15	9.68	18.0	28.9	66.2	199	428	873	1,248	5.19	4.69	3.80	3.19	2.69	2.44	2.13	1.97
1.26	3.86	9.03	16.8	27.9	61.7	186	400	722	1,164	4.52	4.09	3.31	2.78	2.34	2.12	1.86	1.72
1.17	3.68	8.38	15.6	25.9	57.3	172	371	670	1,081	3.90	3.53	2.86	2.40	2.02	1.83	1.60	1.48
1.08	3.40	7.74	14.4	23.9	52.9	159	343	619	998	3.30	3.00	2.43	2.04	1.72	1.56	1.36	1.26
0.99	3.11	7.10	13.2	21.9	48.5	146	314	567	915	2.79	2.52	2.04	1.72	1.45	1.31	1.15	1.06
0.90	2.83	6.45	12.0	20.0	44.1	132	286	516	832	2.31	2.09	1.69	1.42	1.20	1.08	0.949	0.877
0.81	2.55	5.81	10.8	18.0	39.7	119	257	464	748	1.87	1.69	1.37	1.15	0.97	0.878	0.769	0.710
0.72	2.26	5.16	9.6	16.0	35.3	106	228	412	665	1.47	1.33	1.08	0.905	0.762	0.690	0.604	0.558
0.63	1.98	4.52	8.4	14.0	30.9	92.8	200	361	582	1.13	1.02	0.828	0.695	0.586	0.531	0.456	0.429

To find the quantity (pounds) of steam of a given pressure and density that will be discharged through a pipe of a given size and 1,000 ft. long, with a certain drop in pressure: First find the permissible drop in the column under the pressure (Table B) then follow across horizontally to the column under the pipe size (Table A); the amount there found will be the pounds steam discharged per minute.

To find the drop in pressure when a pipe of a given size and 1,000 ft. long is to discharge a certain weight of steam per minute of a given initial pressure and density: Find the quantity discharged in the column under the pipe size (Table A), then follow across horizontally to the column under the pressure and density (in Table B); the amount at this point will be the drop in pressure in the given case.

To find the proper pipe size for a given flow of steam per minute, knowing the initial pressure and density and allowable drop: Find the drop in the column under the pressure (Table B) and follow across horizontally (Table A) to the quantity (weight) nearest to that to be discharged (if not exactly); at the head of the column will be found the proper pipe size.

Since the pressure drop is directly proportional to the pipe length, the drop for any length other than 1,000 ft. may be found by multiplying the drop per 1,000 ft. by the length under consideration and dividing by 1,000.

Templets for Make-Up Connections*

BY F. R. PARSONS

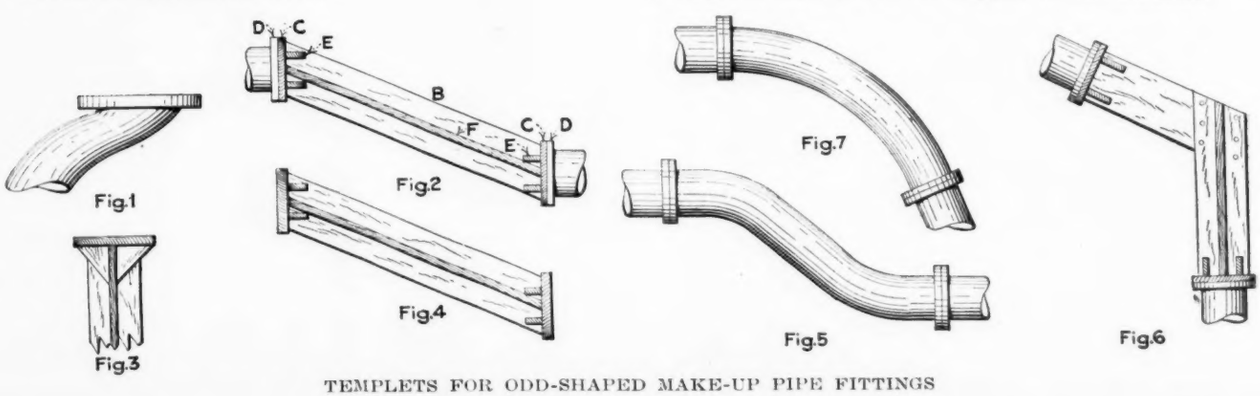
Jointing up pipe lines often presents difficulty, inasmuch as the meeting piece may have to be of intricate shape, or the lines may be out of square, level or line. It is usually best to arrange for whatever is special in the way of connections to be the last consideration and arranged to come in whatever position is most suitable in regard to disposition of the line, possible obstructions and convenience of making up, as well as to construct a templet of the making-up piece in place. Swan-necks, set-offs or bends require to be of easy curves; Y-pieces or breeches pipes are best not too abrupt in their junctions, and a quick curve leaving a flange often makes the question of bolting difficult.

In the construction of templets for making-up pieces, the first thing required will be two wooden flanges of the same thickness and diameter as the existing flanges on the pipe lines. These must be bored and secured to the pipe flanges with two or three bolts. This done, prepare a length of 1-in. stuff, of a width which will for preference come within the bolt holes, and fit it lengthwise from flange to flange, as shown at B in Fig. 2, CC being the two wooden flanges, DD the cast-iron pipe flanges. Now attach this by nails or screws to the flanges by means of the short fillet pieces EE, strengthening it throughout its length by strips, one on either side, reaching from flange to flange, shown at F, this being more clearly indicated in the smaller figure in plan at Fig. 3. Having screwed or nailed everything firmly together, the bolts attaching the templet to the pipe flanges may be removed and the former lifted out of position.

The templet now represents the actual dead length of the required make-up connection, minus whatever might be allowable for jointing. In order to fit the templet for the foundry, it now remains to add the thickening amount to each flange to allow for contraction and machining. This will take the form of extra pieces screwed to the face of each wooden flange, of such a thickness as will be equivalent to the contraction plus whatever amount is allowed for subsequent machining, the finished templet then appearing as in Fig. 4. It does not, of course, imply that the actual pipe will conform to the outlines of the templet, this being used in the foundry only to govern lengths, amount of set-off, angle of flanges and so on. The finished connection will of course be a double bend, or swan-neck, as shown in Fig. 5.

When the casting is made and it comes into the shop for machining, the thickening pieces are removed from the templet in order to furnish a guide to the machinist in respect to length. It is also usual in such cases not to mark off bolt holes in the flanges from the templet, but if it is at all possible temporarily to put the connection into position and mark the holes off from the bolt holes in the flanges of the pipe lines. It is also a distinct advantage whenever possible to set out center or other reference lines through the templets or flanges in order to serve as a guide to the machinist.

*From the "Mechanical World."



TEMPLETS FOR ODD-SHAPED MAKE-UP PIPE FITTINGS

Fig. 6 shows how templets are usually built up and stiffened to suit various angled bends, Fig. 7 showing how the resultant connection would be curved.

Different Electric Systems

In a paper read before a joint meeting of the electrical section of the Western Society of Engineers and the Chicago section of the American Institute of Electrical Engineers, A. J. Goedjen pointed out that, according to a recently published electrical directory of the United States, there are no less than 4,700 central stations in towns of less than 50,000 inhabitants. An analysis of the data relative to 576 systems in six representative states indicates that 15 per cent. use direct current at voltages of 125,-250 and 550 volts two-wire, and 125 to 250 volts three-wire. How well some of these direct-current systems are applied is not apparent, but it is evident that the voltage of direct-current systems is quite well standardized. Of the total number of systems analyzed, 85 per cent. employed alternating current in 22 different combinations of number of phases, wires, cycles and volts, practically none of which are convertible into another combination without great difficulty and expense. In the 488 alternating-current systems analyzed, 24 per cent. were single-phase, 13 per cent. three-wire two-phase, 1 per cent. four-wire two-phase, 39 per cent. three-wire three-phase, and 23 per cent. four-wire three-phase. Of these 488 systems, classified according to the voltages, 2 per cent. operate at 115 volts, 17 per cent. at 1,100 volts, 58 per cent. at 2,200 volts and 24 per cent. at 4,000 volts. The frequencies at which the 488 systems operate show even a greater diversity, 2 per cent. being 25-cycle, 0.6 per cent. 40-cycle, 90 per cent. 60-cycle, 0.2 per cent. 116-cycle, 1 per cent. 125-cycle, and 6.2 per cent. 133-cycle. The 116-, 125- and 133-cycle systems are small plants. It is very apparent from the preceding that there is a great diversity in the fundamental character of the systems in use at the present time.

PERSONALS

J. E. Bridges has been recently appointed meter specialist for the San Francisco office of the Westinghouse Electric and Manufacturing Co.

Herbert F. Henrickson, until recently in charge of large plants in the South, is now chief engineer of the New Cape May Hotel at Cape May, N. J.

E. C. La Rue, United States hydraulic engineer, formerly at Salt Lake City, has been assigned to the special water-power investigation in Oregon.

Edward L. Zeltner, formerly with the C. W. Hunt Co., of New York City, is now with the United States Government in the ordnance department of the navy yard at Washington.

A. K. Webster has resigned from the drafting and designing department of the Illinois Central R.R. to become sales engineer in the Chicago office of the American Radiator Co.

R. H. Rathbun, formerly chief engineer for the Madera Co. at Madera, Chihuahua, Mexico, is now associated with the Chicago branch of the Westinghouse Electric and Manufacturing Co.

Joseph E. Love has resigned as engineer of the South Dakota Railroad Commission and is now connected with the engineering department of the Chicago Union Station Co., Chicago, Ill.

Joseph A. West, formerly chief engineer of the Ogden (Utah) Rapid Transit Co. and of the Ogden, Logan & Idaho Ry., has been appointed chief engineer of the Sumpter Valley R.R. of Oregon.

R. A. Lundquist, consulting engineer, of Minneapolis, Minn., has been selected by the Bureau of Foreign and Domestic Commerce to study the markets for electrical goods in all the big Eastern counties.

Waldo C. Cole, until recently connected with the sales department of the Westinghouse Electric and Manufacturing Co. at San Francisco, has been placed in charge of the publicity department of that office.

W. P. Lyons, formerly vice-president of the Lyons Boiler Works, has been appointed special representative of the H. W. Johns-Manville Co. for the entire Chicago district. He will specialize on the application of J. M. steam and high-temperature cements.

J. B. Long, of the contract department of the Commonwealth Edison Co., Chicago, and previously with the General Electric Co., of Schenectady, N. Y., has been appointed business manager of the Central Power Co., Grand Island, Neb.

Tracy E. Bibbins, for years connected with the Edison General Electric Co., of Portland, Ore., and more recently manager of the San Francisco office of the General Electric Co., has been elected president of the Pacific States Electric Co.

Cecil Hamelin Taylor, until recently consulting engineer of the Republic Motor Truck Co., Alma, Mich., and of the Curtis Aëroplane Co., Hammondsport, N. Y., has gone abroad to study aëronautical developments and gasoline-motor improvements in England and France since the outbreak of the war.

MISCELLANEOUS NEWS

The Walworth Manufacturing Co. moved into its new quarters at 220-222 North Desplaines St., Chicago, Ill., on Sept. 1, where it will have more spacious quarters to handle its large stock of goods.

The Farnsworth Manufacturing Co., of Boston, Mass., manufacturer of steam utilities, has opened a Chicago office at 20 East Jackson Boulevard, with Charles O. Lindahl in charge as managing district engineer.

The Nelson Valve Co., of Philadelphia, has opened an office in the Hearst Building, Chicago. H. J. Moyer has been appointed to represent both the Nelson Valve Co. and Yarnall-Waring Co. in this territory.

The Smoke-Abatement Ordinance which has been under consideration in Lexington, Ky., has been drafted and has undergone a first reading before the city commission. Numerous small manufacturers and proprietor of steam plants of various kinds were present and voiced their protests against the terms of the measure, which would virtually compel installation of smoke-consuming and automatic-stoking equipment, it is stated. At a second reading numerous amendments are expected to be offered.

Recent Court Decisions

Digested by A. L. H. STREET

Franchise Rights of Electric Companies—An electric company holding a municipal franchise to manufacture and sell electric current for lighting and power purposes is entitled to purchase the necessary current from another company, in the absence of a special restriction against it. (Texas Court of Civil Appeals, City of Terrell vs. Terrell Electric Light Co., 187 "Southwestern Reporter," 966.)

Power Appliances as Removable Fixtures—Motors installed by the lessee of a building to operate an elevator, and other appliances put in at his own expense, may be removed by him when the removal can be effected without injury to the premises and when there is no agreement restricting his right to take them away. (Michigan Supreme Court, Lovett vs. Bermingham-Seaman-Patrick Co., 158 "Northwestern Reporter," 881.)

Validity of Occupation-Tax Ordinance—An ordinance of the City of Lincoln, Neb., that purported to levy an occupation tax on a public-service company that furnishes gas for power and other purposes is invalid for unjust discrimination when no such tax is levied against other companies in the same city furnishing electricity for like purposes. (Nebraska Supreme Court, City of Lincoln vs. Lincoln Gas and Electric Light Co., 158 "Northwestern Reporter," 962.)

Violation of the Sunday Laws—Operation of a power plant on which public water supply or lighting depends is such work of necessity as is exempted from the provisions of the laws that forbid unnecessary labor on the Sabbath, but work in providing fuel for a power plant is unlawful when done on Sunday and when there is nothing to show that ample fuel could not have been provided on secular days. (Arkansas Supreme Court, Wilson vs. State, 187 "Southwestern Reporter," 937.)

Contributory Negligence of Lineman—There can be no recovery for death of an electric lineman who, while placing a converter in place in the installation of electrical appliances

in a power plant, permitted his head to come in contact with a switch that was charged with an electric current of 13,000 volts; he having known of the existing conditions as shown by his participation in the placing of a sign at that point showing the voltage. (Maryland Court of Appeals, Westinghouse Electric and Manufacturing Co. vs. State, 98 "Atlantic Reporter," 206.)

Contributory Negligence of Employee—In dismissing a suit brought by a former employee whose hand was crushed in cogwheels while he was cleaning and soaping a pulley belt without stopping the machinery, the South Carolina Supreme Court holds that he must be deemed to have assumed the risk of the accident by failing to stop the machinery, which could have been readily done; it appearing that he was experienced in such work and that there were no hidden dangers. "A servant must exercise common sense in obeying the reasonable orders of his superior, if he knows it is dangerous and that it is likely to end in injury to himself; then his primitive instincts should forbid his doing it." (Cordele vs. Hampton Cotton Mills, 89 "Southeastern Reporter," 498.)

NEW PUBLICATIONS

ELECTRICAL TABLES AND ENGINEERING DATA. By Henry C. Horstmann and Victor H. Tousley. Published by F. J. Drake Co., 1916; Chicago, Ill. Cloth; 4½x6½ in., 331 pages; 34 illustrations; 82 tables. Price, $1.

This book is a collection of useful tables and practical hints for electricians, foremen, salesmen, estimators, contractors, architects and engineers. The work contains no theoretical discussion, its scope being limited to practical information that is daily called for by those engaged in the electrical industry. The material is arranged in alphabetical order, and the tables are carefully indexed, which greatly facilitates finding any desired item. The section devoted to a discussion on direct- and alternating-current motors is disappointing as the material is not only poorly arranged but in many cases does not represent modern practice. Taken as a whole, however, the book contains a great deal of useful material for the practical man.

ELECTRIC MOTORS—By David Penn Moreton, B. S., E. E. Published by F. J. Drake & Co., Chicago, Ill., 1916. Size 4½x6½ in.; 241 pages; 115 illustrations. Price, $1.

This work is intended to be a practical book for the practical man, dealing with the principles, construction, operation and maintenance of direct- and alternating-current motors. The first three chapters treat of the fundamental principles of electricity and magnetism; direct- and alternating-current circuits. It is the author's purpose to give the reader an understanding of these elementary principles before applying them to the different types of direct- and alternating-current motors. Chapter 4 gives the common method of making electrical measurements that are in general use by the practical man. Chapters 5 and 9 indicate some of the elements of armature winding. Chapters 6, 7 and 8 are devoted to the treatment of the different commercial types of direct-current motors, the various methods of speed control, care and operation, location and repair of direct-current motor troubles. The treatment of the different commercial types of alternating-current motors, including the method of starting, speed control, operating characteristics, care and operation, is given in the last three chapters. This little book should find a field of usefulness among the practical men who have only a limited knowledge of the fundamental principles of electrical machinery.

TRADE CATALOGS

Kerosene Oil Engine. The Burkett Mfg. Co., 162 West Spring St., Columbus, Ohio. Catalog. Pp. 16, 6x9 in.; illustrated.

Indicating and Recording Thermometers. The Foxboro Co., Foxboro, Mass. Bulletin No. 104. Pp. 56; 8x10½ in.; illustrated.

Turbine Blowers. L. J. Wing Mfg. Co., 352-362 W. 13th St., New York. Bulletin No. 37, pp. 22, 6½x9 in. This bulletin superseding Bulletin No. 27 contains considerable data on mechanical draft.

Boiler Tube Cleaners. The Lagonda Mfg. Co., Springfield, Ohio. Catalog L-9. Pp. 36, 7½x10 in. Describes water, air and steam driven tube cleaners and other specialties including strainers, cut-off valves, etc.

"Water Purification for all Purposes." Wm. B. Scaife & Sons Co., Pittsburgh, Penn. This is the title of a treatise on the purification of water with the Scaife and We-Fu-Go systems. Includes analysis from various water supplies in the United States and miscellaneous data for calculating water requirements for various uses.

Proposed Construction

Ala., Birmingham—Alabama Power Co. plans to spend $1,000,000 on extending transmission lines and building substations. W. N. Walmesley, Gen. Mgr.

Ariz., Phoenix—City votes Oct. 3 on bonds for electric-light and power plant and gas plant.

Ark., Waldron—Waldron Power and Ice Co. plans electric-light plant.

Fla., Miami—City sold $25,000 bonds, part of proceeds for electric-light plant. H. E. Elrod, Dallas, Engr.

Ill., Geneseo—City voted $48,000 bonds for electric-light plant. B. S. Gray, City Atty.

Iowa, Des Moines—Des Moines City Railway Co. soon to let contract for rebuilding power station.

Md., Towson—State Normal School plans to install electric-light plant.

Mass., Great Barrington—Great Barrington Electric-Light Co. let contract hydro-electric plant to FISKE-CARTER CONSTRUCTION CO., Worcester.

Mich., Grand Rapids—City plans 3 mi. 6,000-volt transmission line and 200,000-gal. pump in electric-light plant. I. R. Ellison, City Mgr.

Mich., Hamburg—L. M. McCleary, Gregory, plans electric-lighting system in Hamburg.

Minn., Hendrum—Village Council plans electric-lighting system.

Minn., Hibbing—City making plans for power plant. C. L. Pillsbury Co., Metropolitan Bldg., Minneapolis, Engr.

Minn., Hoffman—Village receiving bids Sept. 20 for electric transmission line and distribution system. E. D. Jackson, St. Paul, Consult. Engr.

Minn., Pipestone—Northwestern Light and Power Co. plans transmision line to supply electricity to Edgerton, Woodstock, Jasper and Trosky.

Mont., Columbia Falls—Montana Power Co., Butte, plans addition to power house at Columbia Falls. About $500,000.

Neb., Omaha—Omaha Electric Light and Power Co. let contract addition to power plant to JARRETT-CHAMBERS CO., 30 East 42nd St., New York. About $1,000,000.

N. M., Mogollon—Mogollon Power and Lumber Co. let contract hydro-electric plant to PELTON WATER WHEEL CO.

N. Y., Ogdensburg—Aluminum Co. secured right-of-way for power transmission line to Ogdensburg.

N. Y., Saratoga Springs—G. P. Gunther, Arch., making plans for 2-story power house in connection with school for Redemptorist Fathers, St. Clements College.

N. D., Munich—O. J. Anthony and associates plan electric-light plant.

Ohio, Celina—Board Trustees Pub. Affairs receiving bids Sept. 26 for 300 hp. capacity steam engine at 100 lb. steam pressure, speed about 200 r.p.m. and not exceeding 2 lb. back pressure. G. A. Petrie, Clk. E. S. Kay, Winchester, Ky., Consult. Engr.

Okla., Wayne—City Council plans electric-light plant.

Ont., Elmwood—Village Council plans hydro-electric power and lighting system.

Ont., London—A. Beck, London, Chn. Ontario Hydro Comn., making plans for transmission lines and power stations. About $2,500,000.

Penn., Allentown—Bascom & Sieger, Engr., submitted plans to Council for electric-light plant.

Penn., Beaver Falls—Duquesne Light Co., Pittsburgh, purchased plant of Beaver County Light Co. and plans to improve same. About $1,000,000. F. Vhlenhant, Jr., Pittsburgh, Ch. Engr.

Penn., Philadelphia—Hardwick & Magee Co. plans 1-story boiler plant on 7th St.

Penn., Philadelphia—Pennsylvania R.R. let contract power house at Reed St. yards to ROYDHOUSE-AREY CO., $22,000.

Penn., Pittsburgh—Pressed Steel Car Co. granted permit for 1-story power house on Bison St. About $4,500.

Penn., Reading—Reading Transit and Light Co. plans high tension power line to Kutztown.

R. I., Providence—Wanskuck Co., Branch Ave., let contract 2 additions to boiler house to GUARANTEE CONSTRUCTION CO., New York.

S. C., Spartanburg—South Carolina Light, Power and Railways Co. plans dam, 600-ft. long and 44-ft. high, at Gaston Shoals on Brod River. Electricity to be generated for transmission to Blackburg, Spartanburg, Gaffney, Cowpens, Woodruff and Pacelet.

S. D., Garretson—City voted $18,000 bonds for enlarging electric-light plant.

S. D., Worthing—W. F. Donahue, Worthing, granted franchise by Council to construct electric-light plant.

Tex., Denison—Texas Power and Light Co., Dallas, plans power plant on Red River near Denison. About $400,000.

Wis., Broadhead—Water and Lighting Comn. plans to build power plant or erect new one.